PRELIMINARY DESIGN AND 2D SIMULATION OF SELF-RECTIFYING TURBINES FOR APPLICATION IN THE CHANNEL TUNNEL

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ABSTRACT

The Channel Tunnel is made up of 2 long railway tunnels of about 50 km long and connected every 250 m by 2 m diameter pipes. These pipes, called piston relief ducts, allow the air pushed by a train to go from one tunnel to another to limit the piston effect due to the trains. Pressure drop devices are placed in the ducts to limit the flow speed. Eurotunnel is studying the possibility to replace these devices with turbines. These turbines must operate with flows alternatively in both directions, while keeping the same rotation direction. This study focused on the design and simulation of self-rectifying turbines adapted to the Channel Tunnel ducts. It is therefore proposed here to evaluate the adaptation of the various existing concepts by means of one-dimensional calculations and 2D numerical simulations. Two types of turbines are studied: Wells and Impulse turbines. A preliminary design of the two types of turbines is proposed and their performances are discussed.

KEYWORDS : Self-Rectifying Turbine, Design, Channel Tunnel.

NOMENCLATURE :

Roman letters

c chord length [m]
C_x axial force coefficient $F_x/(1/2\,\rho w^2 c)$
C_y tangential force coefficient $F_y/(1/2\,\rho w^2 c)$
F_x axial force [N]
F_y tangential force [N]
L PRD length [m]
nb number of blades
P power [W]
Q volume flow rate $[m^3/s]$
r radius [m]
t time [s]
u blade velocity [m/s]
v absolute velocity [m/s]
v_t tangential velocity component [m/s]
v_x axial velocity component [m/s]
w relative velocity [m/s]
p static pressure [Pa]
p_t total pressure [Pa]

Greek letters

Δp static pressure drop [Pa]
Δp_t total pressure drop [Pa]
η efficiency $(p_{t\_in} - p_{t\_out})/(p_{t\_in} - p_{out,is})$
ξ friction loss coefficient [-]
ρ density $[kg/m^3]$
σ solidity $nb \cdot c/(2\pi r)$
φ flow coefficient (v_x/u)
ω angular velocity [rad/s]

INTRODUCTION

The Channel tunnel is made of two 50 km long tunnels connecting France and England, and 195 transversal tunnels (almost one every 250 m) called pressure relief ducts (PRD) with a 2 m diameter and 20 m long. These PRD were built to equalize the pressures between the two long tunnels and to reduce the piston effect which is created by the trains moving up to 160 km/h. The PRD are at the origin of the problem which arises when two trains are crossing the same PRD section. In this case, the flow established in the PRD creates some dangerous side forces on the trains. Pressure drop devices, called 'flow restrictors', were therefore installed to create a pressure drop and to reduce the flow velocity in the PRD.

The Channel tunnel total consumption of energy in one year is about 500 GWh, 30 GWh of which (6%) is used for refrigerating systems that keep the temperature at 25° C to control the heat mainly due to aerodynamic friction. A fraction of this heat load is caused by friction on the flow restrictors that are present in the PRD. Eurotunnel has therefore considered the possibility of replacing the flow restrictors with turbines that would produce the same pressure drop but recover a part of the kinetic energy of the flow to produce electrical energy. There are two advantages: there is energy sparing for refrigerating systems and at the same time a production of electrical energy. A first estimate of the power dissipated on the restrictors can be deducted considering the velocity and the pressure drop characteristics shown in figure 1 and obtained from measurements. These measurements have been made by Eurotunnel during a period of 110 min in the most frequented hours.

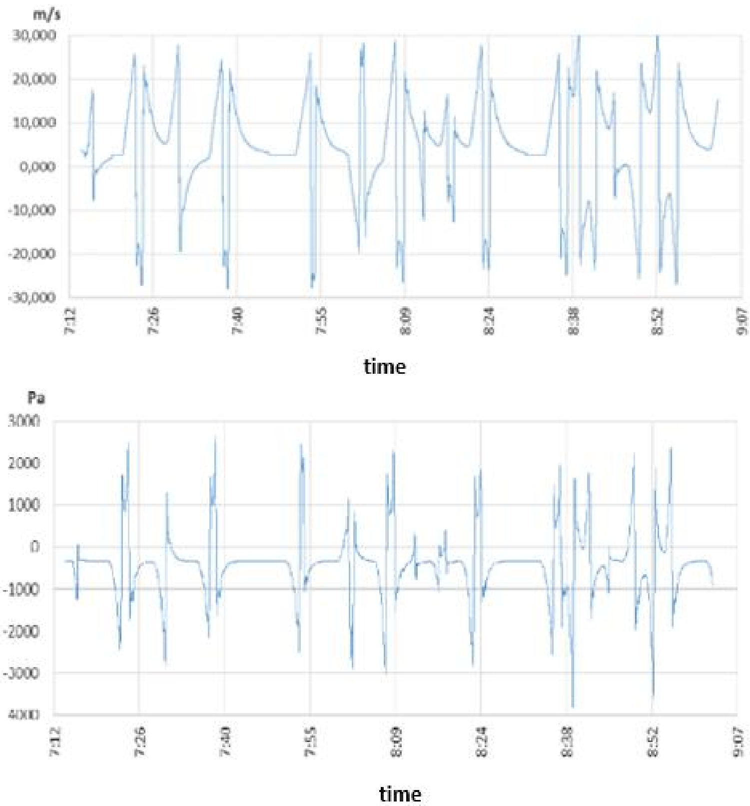

Figure 1: Instantaneous velocity and pressure drop measured in the PRD (Division Infrastructure Eurotunnel 2010)

With the help of Weibull distribution of the velocity data in figure 1, it was estimated that 5.7 GWh per year of energy are dissipated by the flow restrictors. The aim of the present project is to try to recover a part of this energy.

For this application, self-rectifying turbines (figure.2) are needed, since they keep the rotation in one single direction, irrespective of the axial flow direction, thus facilitating turbine operation and mechanical connection to electrical machines. The Wells turbine without guide vanes and the Impulse turbine, both traditionally used for OWC (Oscillating Water Columns) plants were considered in this study.

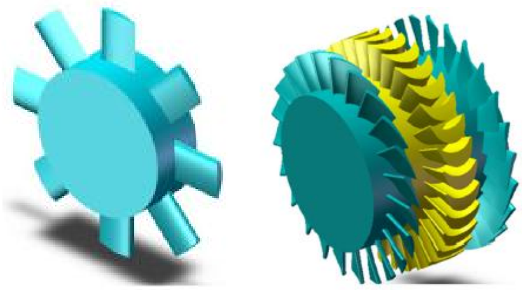

Figure 2: self-rectifying turbines: on the left Wells turbine, on the right Impulse turbine from[www.scoop.it/turbines/p/4007344956/2013/09/09/numerical-analysis-of-impulse-turbine-for-isolated-pilot-owc-system]

A comprehensive review of OWC technology is given in Falcao and Henriques (2016) and Setoguchi and Takao (2006), the second focusing especially on the types of air turbines used. Following the decision of the European Commission in 1991 to include wave energy in their R&D program of renewable energies, two pilot plants were built on Pico Island, Azores, Portugal,(400 kW) and LIMPET project on Islay Island, Scotland (500 kW). Both power plants have been equipped with Wells turbines, invented in 1976 by Dr Alan Arthur Wells (G.B. Patent No. GB1595700, 1976). A reference study for the operation and analysis of Wells turbine is given in Raghunatan, (1995). Several other studies (experimental in Camporeale et al. 2011**)**, numerical (Ghisu et al. 2015), or joint experimental/CFD (Taha et al. 2010) are available in literature.

The alternative to the Wells turbine is the self-rectifying Impulse turbine patented by Ivan A.Babintsev in 1975 (U.S. Patent No. US3922739, 1975), tested in the Trivandrum 125 kW power plant in India. This type of turbine has been experimentally tested (Setoguchi et al. 2001) and compared with the Wells turbine. Both unsteady and steady cases have been analysed (Zheng Liu et al. 2017). According to the studies done until today, it is known that the main limitation of the Wells turbine is the occurrence of stall which limits the range of operativity. The latter, on the other hand, suffers a lower peak efficiency because of the incidence losses due to the symmetry constraint on the geometry.

In the present study, the preliminary design of the two turbines adapted to the Channel Tunnel is discussed : a mean radius 1D model , and 2D cascade simulation with the CFD code Star-CCM+ were developed, and a comparison between the two turbines was made in terms of efficiency and operating range.

DESIGN DATA

In the present study, only steady state calculations were done, leaving unsteady calculations for a future level of analysis; steady calculation already highlights many of the salient operating features of the two types of turbines especially for a preliminary comparison and provides characteristic curves that can be used in quasi-steady analysis as a first approximation to tackle the unsteady problem. The steady calculations are based on a definite value of velocity and pressure drop taken as design point values. The choice of the two values is discussed in the following: since the turbines have to replace the flow restrictors, the aim is that they have similar behavior in terms of pressure drop- velocity characteristics.

The instantaneous pressure drop in the PRD consists of a friction term, and an inertial one.

$$\Delta p = \xi \cdot \frac{1}{2}\rho v^2 \ + \ \rho L \frac{dv}{dt} \quad (1)$$

It was seen that the inertial term can be neglected and according to Eurotunnel analysis the restrictor can be characterized by a loss coefficient of 4.2 (Division Infrastructure Eurotunnel 2010).

The design values of pressure drop and velocity for the turbines have been therefore chosen so that the ratio

$$\frac{\Delta p}{\frac{1}{2} \cdot \rho \cdot v^2} \quad (2)$$

is equal to the restrictor loss coefficient. The design velocity has been chosen to be 20 m/s because according to an energy production calculation done by Eurotunnel (Sabatier 2015) , this is the velocity for maximum energy density. The design pressure drop is therefore fixed at 1000 Pa +/- 5%. The external diameter of the turbine is fixed at a maximum value of 2m by the PRD geometry constraint and the hub to tip ratio is fixed at 0.5, a reasonable value for self-rectifying turbines (Falcão and Gato 2012).

WELLS TURBINE

For the preliminary design, a 1D model has been developed using blade element theory (Raghunatan, 1995). The most relevant quantities calculated by the code are power, pressure drop, and efficiency. The code calculates the forces on the blades using lift and drag coefficients for an airfoil in free stream (Airfoiltools.com 2018), (Timmer 2010) and using corrective coefficients for axial and tangential forces C_x and C_y found in Raghunatan (1995) to take into account effect of solidity. The code inputs are the design point axial velocity and the number of blades.

To find which geometrical characteristics of the turbine are best suited to match the desired pressure drop at the design velocity to have highest efficiency, the code imposes various combinations of flow coefficient and solidity and for each combination it calculates power, pressure drop and the efficiency of the machine. A matrix is created, in which for each combination of flow coefficient and solidity, the values of pressure drop and efficiency are stored. The solutions for which the pressure drop is outside the required range are not considered. Then, by comparing all the other solutions, the one that gives the maximum efficiency is chosen to be the optimal one. The input number of blades is then changed, to see which number of blades is the optimal one.

To validate the model, 2D simulations at the mean radius have been made in Star CCM+. The (k-ω) turbulence model has been selected. The geometric domain is represented in figure 3.

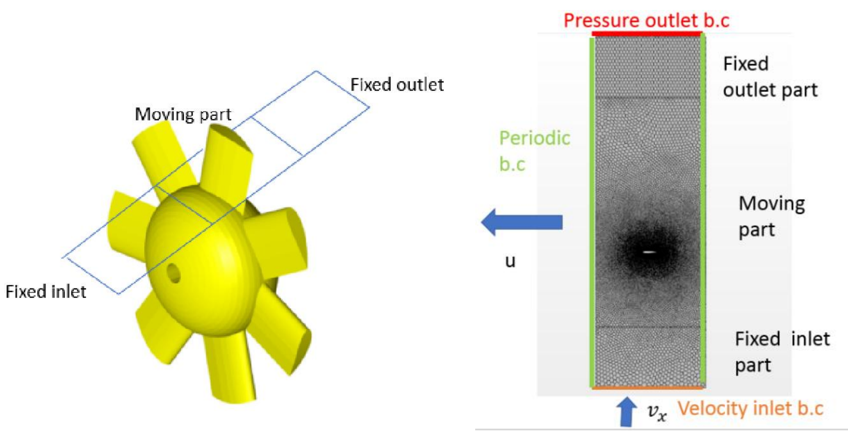

Figure 3: geometric domain for CFD simulation and the mesh used with its boundary conditions

The mesh has been built using a polygonal mesher with base size equal to the chord length, target size of 10% and minimum target size of 5%. The surface growth rate is 1,02 and the aspect ratio outside the boundary layer is between 0.9 and 1. Around the blades a prism layer mesher with 20 prism layers is used, with a prism layer total thickness of 1%, and a stretching of 1,1. The aspect ratio in the boundary layer gradually reduces until a value of 0.09 on the closest to the surface layer. To further increase the detail around the blades a surface control was built which reduces the target surface size to 0.5% around the blades, with a surface growth rate of 1,01. The resulting number of cells is of the order of 60000.

The flow around one blade has been simulated with periodic conditions. A frozen rotor calculation has been used. The total domain has been divided in three sub-domains, fixed regions upstream and downstream, and a central region in a moving reference frame at the blade velocity. An axial velocity of 20 m/s is imposed at inlet; and a relative pressure equal to 0 Pa is imposed at the outlet. The sides of the three regions are modelled with a periodic boundary condition.

For the specific case of a turbine with 8 blades, the comparison between 1D and CFD results is shown in figure 4: the dashed lines represent 1D results, while continuous curves represent CFD. The 1D model predicts a lower total pressure drop and thus predicts a higher efficiency. The reason for the discrepancy could lie in the fact that 1D code uses corrected coefficients - taking into account solidity effect- only for the axial and tangential force, but not on the drag coefficient which is used to calculate the total pressure drop. With an increase in solidity, the drag coefficient increases and thus the total pressure drop.

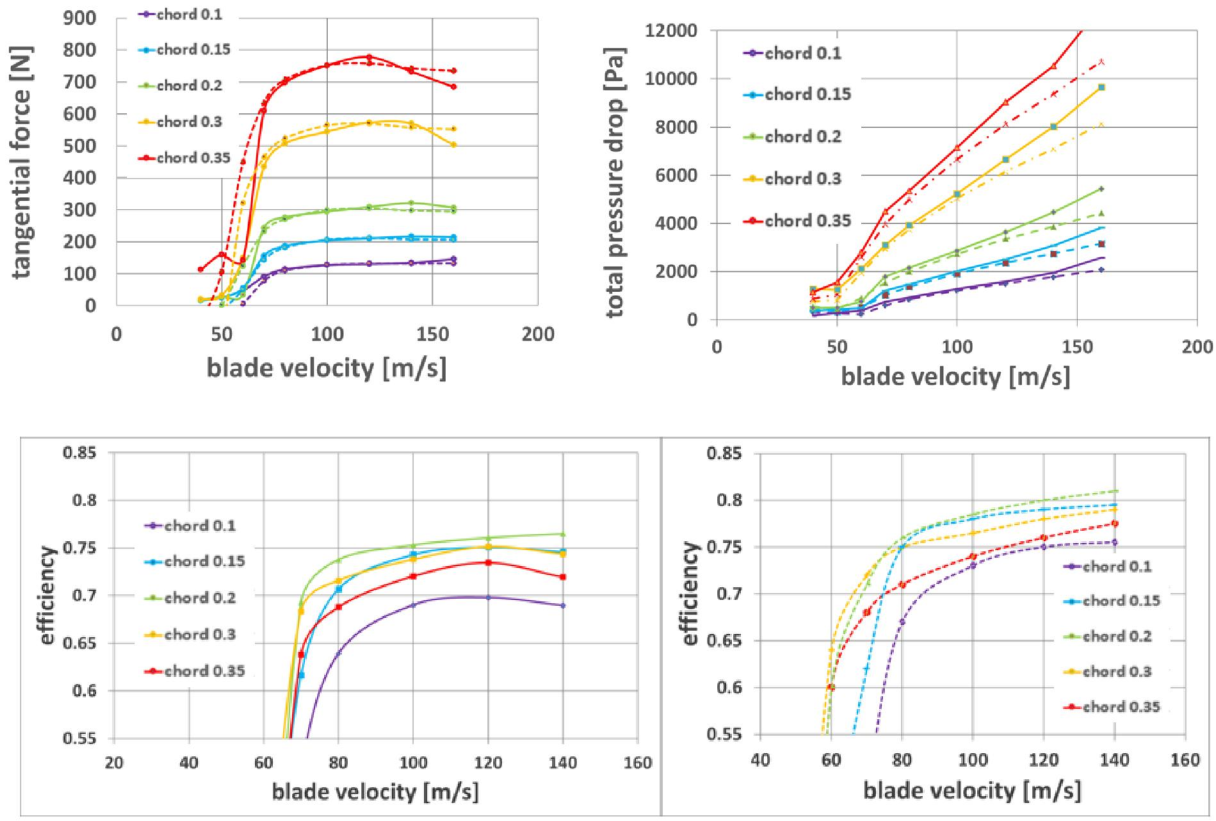

Figure 4: tangential force, total pressure drop and efficiency of the 8 blades Wells turbine according to 1D (dashed curves) and CFD models (continuous curves)

The 1D code was then tested with different numbers of blades to see which number of blades and chord length assure the highest efficiency stated that the pressure drop is 1000 Pa +/- 5%. The design blade velocity was chosen to be 80 m/s, since for lower velocities the stall limit is approached as can be seen in figure 4. The strong decrease of efficiency for blade velocities around 60/70 m/s is due to stall, since the corresponding angle of attack is bigger than 20°, in those cases.

Higher blade velocities are not also good option, because they would lead to higher values of some local Mach numbers, especially at blade tip which could lead to a transonic regime. In figure 5, the best turbine designs for each number of blades that satisfy the required pressure drop are represented. Each point represents a specific turbine with number of blades specified in the x axis. The results of the 1D calculation leads to a turbine with low solidities of around 0.2. This is due to the fact that the pressure drop depends on solidity as a power of 1.6 (Raghunatan 1995). Consequently, limiting the pressure drop to 1000 Pa limits the solidity of the machine (figure 4). The efficiencies are higher for lower number of blades configurations. The solutions with one and two blades were excluded since they lead to high chord lengths. The solutions with 3 and 4 blades were tested numerically and finally the solution with 3 blades was chosen since it has higher efficiency. It can be noted that this is an unusually small number of blades, compared with Wells Turbines used in OWC applications. This is due to the fact that, in the present case, the pressure drop has to be limited to 1000 Pa.

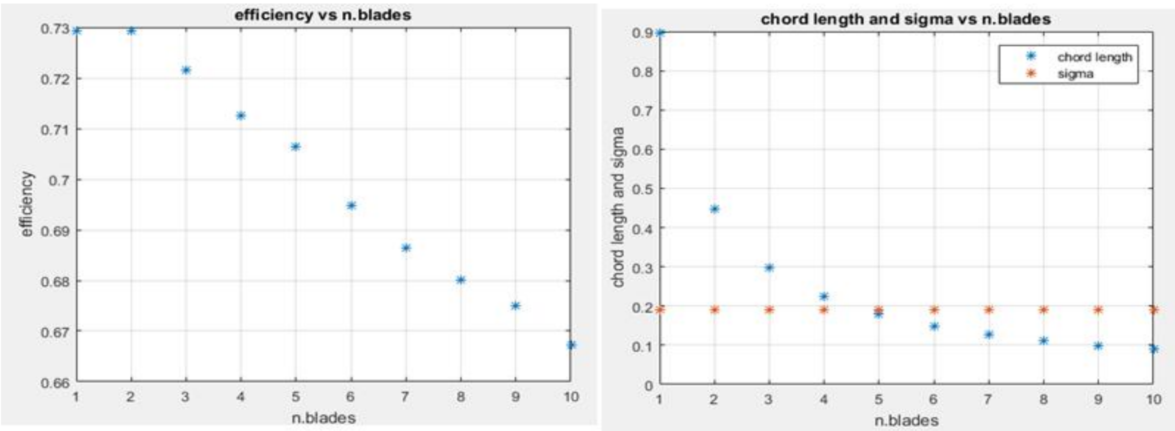

Figure 5 : 1D optimal solution for different number of blades. Efficiency (left figure), Chord length and Solidity (right figure) as a function of the number of blades

The final parameters of the Wells turbine are presented in table 1:

Table 1 : Design parameters for the Wells Turbine

Solidity	0.19	Power (kW)	40.2
Number of Blades	3	Total Pressure Drop (Pa)	1007
Chord length (m)	0.3	Pressure drop (Pa)	1040
u (m/s)	80	Efficiency	0.675
φ	0.25		

An energy production calculation was done for this turbine from the power curve and the Weibull distribution of the velocities shown in figure 1. Only 150 turbines were considered to take into account entry and exit effects in the tunnels that reduce power extracted, and 12 hours per day of operation were considered. The energy produced in one year is of 4.65 GWh (17.6% of the value that would be extracted if the turbine could always operate at nominal power at all flow conditions).

In figure 6, the turbine pressure drop is plotted, in the case of constant rotational velocity and in the case of variable rotational speed in order to keep constant the flow coefficient. In the case of a constant rotational speed the blade stalling results in a sudden pressure drop decrease for axial velocities higher than 25 m/s. If the rotational speed is varied starting from the design point in order to keep a constant flow coefficient, the pressure drop curve matches the curve of the restrictors, because, in this case, the turbine is supposed to always be at operating points in similarity with the design point.

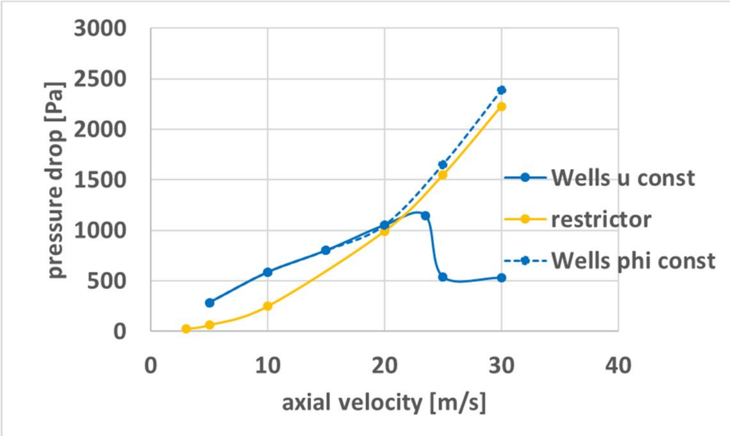

Figure 6: pressure drop as a function of axial velocity for Wells turbine compared to the restrictor

However, some CFD calculations have shown that under variable rotational speed, the Mach number can reach values of 1 at the blade tip for axial flow velocities higher than 25 m/s, a condition incompatible with the application of Eurotunnel, especially because of noise production.

IMPULSE TURBINE

For the preliminary design, a 1D model has been developed. The model performs an analysis at the mean radius building velocity triangles and using some classical correlations for the total pressure drop: Ainley & Mathieson (1955) for inlet stator and rotor, Howell (1945) for the outlet stator in which the flow is decelerated and thus acting as a diffusing row. The code is built to find the best configuration especially in terms of blade angles. It is known that the main disadvantage of Impulse self-rectifying turbines is the strong incidence losses that arise because of the symmetry constraint. These losses strongly depend on the blade angles.

In figure 7 the mean section of the turbine is shown, and two extreme cases are represented: in the right figure the rotor blade angle is chosen to be such that, at design point, no incidence losses are present at the rotor inlet. But strong incidence losses occur at the inlet of the second symmetric stator. If the rotor blade angle is chosen to be such as no incidence losses are present at the second stator (case on the left in figure 7), there will be strong incidence losses at the inlet of the rotor. It is thought that there must exist a choice of blade angles for which the incidence losses are minimized and efficiency maximized.

To choose the optimum blade angles, the 1D code is structured in the following way:

- A function that takes as input the rotor blade angle and calculates the values of stator blade angle for different values of flow coefficient, in order to satisfy the required pressure drop. For each flow coefficient power and efficiency are also calculated. There will be one flow coefficient that will maximize efficiency.
- A main program that for different values of rotor blade angle calls the function to give the optimal flow coefficient and the corresponding efficiency for each rotor blade angle. The main program, then compares the efficiency to find which rotor blade angle is the best one.

The code performs calculations for an imposed value of chord length and calculates the optimal number of blades using Zweifel correlation. The output of the main program is the plot in figure 8, in which are represented the efficiency for each blade rotor angle (measured from the meridional direction) and the corresponding flow coefficient for which this efficiency is found. The chord length was imposed to be 0.3 m.

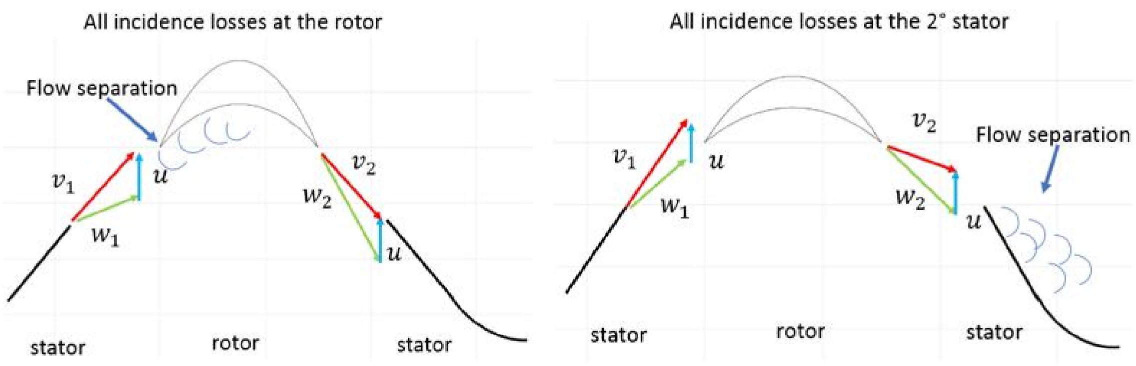

Figure 7: two extreme angles configurations for impulse turbine

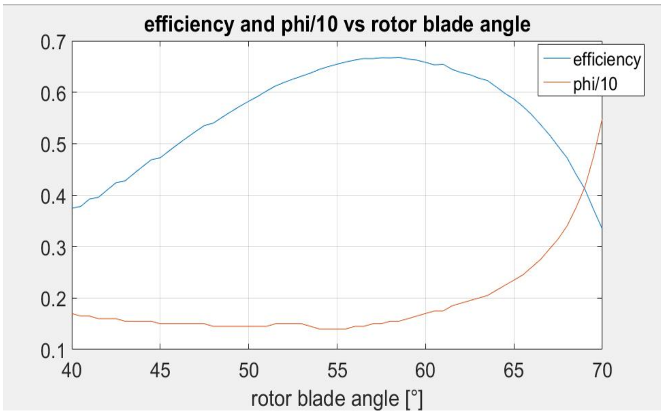

Figure 8 : efficiency and flow coefficient/10 as a function of rotor blade angle (measured from the axial direction)

It was found that both the blade rotor and stator angles are close to 60°, which corresponds to the situation in which velocity triangles are symmetric; this physically implies distributing equally the incidence loss between rotor and second stator. This value of 60° is also the value obtained by Setoguchi et al. (2001) which have done experiments on a model of Impulse turbine.

As in the case of the Wells turbine, a CFD model was built in Star CCM+ and run for different rotational velocities and chord lengths. The 2D model was built similarly to the one used for the Wells Turbine. A frozen rotor approach with inlet velocity, outlet pressure and periodic boundary conditions are adopted. The mesh is represented in figure 9 and it was built with the same criteria described for Wells turbine. The number of cells is of the order of 50000.

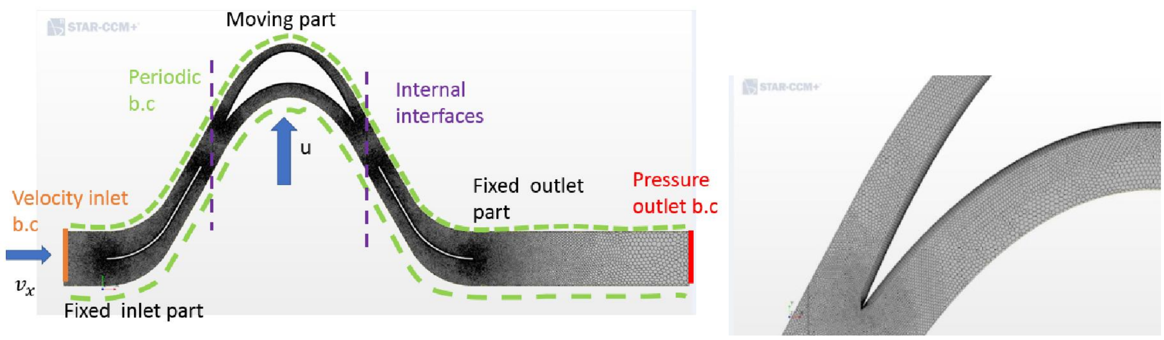

Figure 9 : geometry and mesh for the impulse turbine CFD simulations

In figure 10 the results of 1D code and CFD calculations are shown for a turbine with both stator exit blade angle and rotor blade angle equal to 60° and 40 blades.

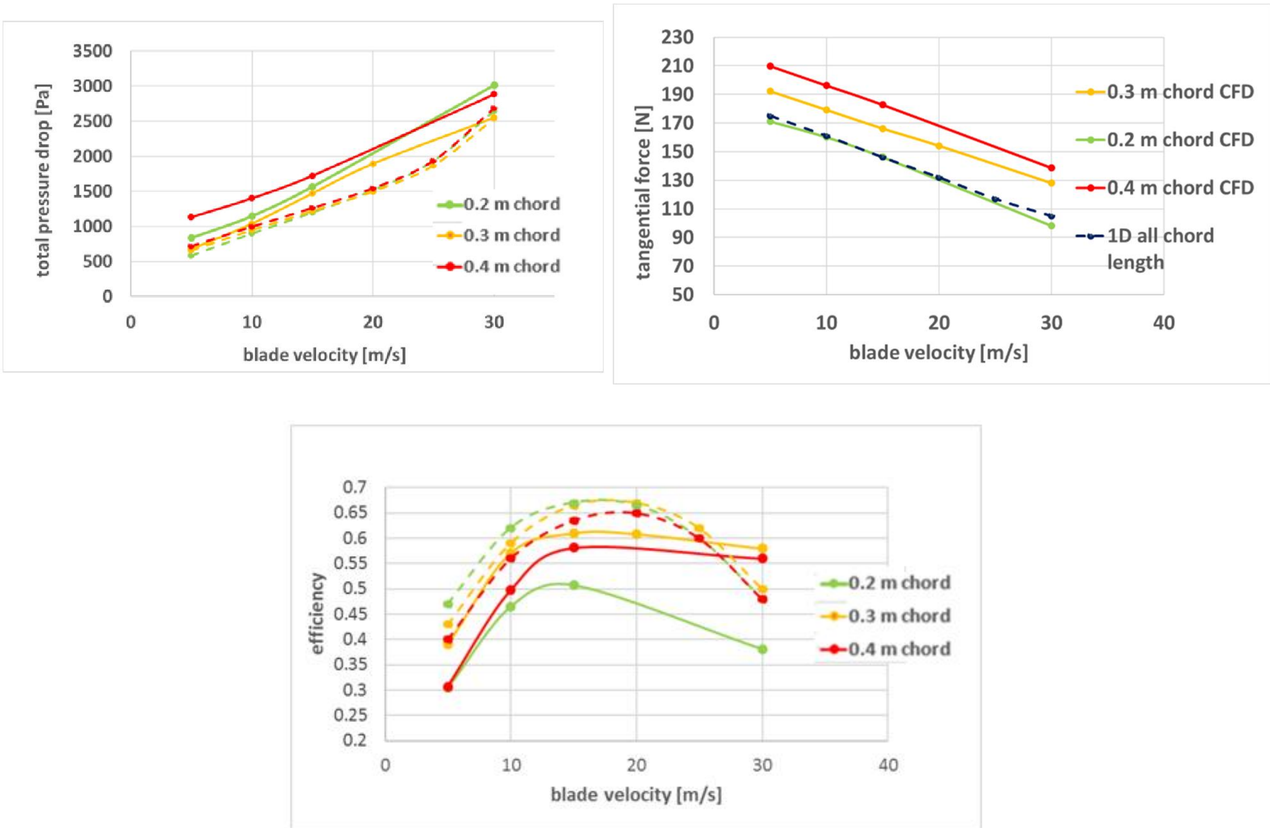

Figure 10 : total pressure drop, tangential force , efficiency for a 40 blades impulse turbine according to 1D Code (dashed lines) and CFD (continuous lines)

It was seen that because of lower kinetic energy in the wake compared to Wells turbine, the total pressure drop is more than 95% of the static pressure drop. In the 1D code the effect of solidity on tangential forces wasn't taken into account because of lack of correcting coefficients so in figure 10 there is only one curve for the 1D simulation.

The 1D code predicts an almost linear growth of the pressure drop with rotational velocity in the range of pressure studied in the present paper**.** Like what was observed with Wells Turbine CFD calculations predict higher pressure drops which reflect in a lower efficiency compared to 1D results. The main limitation of the 1D code is the poor sensibility to modifications of solidity: according to CFD there is a value of chord length of 0.3 m that minimizes the pressure drop and maximizes efficiency. Lower and higher chord lengths give rise to higher pressure drops.

Therefore, a study of the effect of solidity was done using CFD: a design rotational velocity was fixed at 9 m/s (115 rpm) and different combinations of chord length and number of blades were tested (figure 11).

chord length [m] / N° blades

solidity	0.2	0.3	0.4
30	1.27	1.91	2.55
40	1.70	2.55	3.40
50	2.12	3.18	
60	2.55	3.82	

chord length [m] / N° blades

Δp [Pa]	0.2	0.3	0.4
30	1234	1047	1130
40	1110	1030	1343
50	1039	1260	
60	1076	1474	

		chord length [m]		
	force [N]	0.2	0.3	0.4
N° blades	30	149	212	238
	40	152	182	199
	50	134	148	
	60	115	117	

		chord length [m]		
	efficiency	0,2	0,3	0,4
N° blades	30	0,30	0,49	0,53
	40	0,44	0,55	0,48
	50	0,52	0,49	
	60	0,53	0,40	

Figure 11 : values of, pressure drop, tangential force and efficiency for different values of solidity of a turbine running at 115 rpm

It was seen that there is a narrow range of solidity that minimizes the pressure drop: low solidities give rise to flow detachment while high solidities make the pressure field around the rotor blade more uniform so that an important decrease in tangential force is observed. There is thus a solidity which gives highest efficiency corresponding to 40 blades and 0.3 m chord length. The design data for Impulse turbine are represented in Table 2.

Table 2 : design parameters for Impulse turbine

Solidity	2.55	Power (kW)	32.7
Number of Blades	40	Total Pressure Drop (Pa)	980
Chord length (m)	0.3	Pressure drop (Pa)	1030
u (m/s)	9	Efficiency	0.55
φ	2.22		

An energy production calculation was done using Weibull distribution and the Impulse power curve obtained from CFD which gives an annual production of 4.55 GWh, (21.6% of the value that would be extracted if the turbine could always operate at nominal power at all flow conditions). The loading factor (0.216) is higher than for Wells turbine (0.176) so even if peak efficiency is lower for Impulse, the energy extracted is similar. In figure 12 the pressure drop developed by the turbine is represented, in the case of constant rotational velocity and in the case of variable rotational speed in order to keep constant the flow coefficient starting from design point.

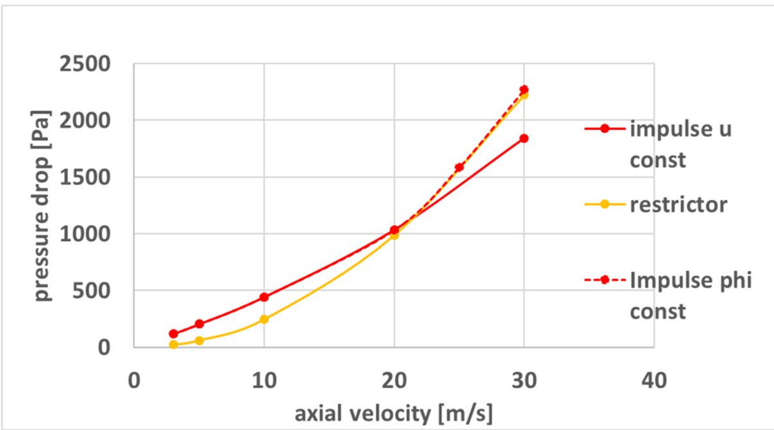

Figure 12 : pressure drop of impulse turbine compared to the restrictor

Contrary to the Wells turbine, if a speed control is realized to keep constant the flow coefficient, no problems related to the Mach number appear, since Mach number values are generally less than 0.3 as some CFD calculations have shown. The energy extracted under variable speed control could be slightly higher: an annual production of 4.95 GWh could be expected.

COMPARISON OF WELLS AND IMPULSE TURBINE OPTIMAL DESIGN

A comparison was finally done between the two types of machine: in figure 13, the efficiency and pressure drop are represented as a function of the axial velocity. The curves were obtained keeping rotational velocity constant and varying the axial velocity. The restrictor pressure drop is also plotted. What appears clearly is that impulse turbine offers a wider operating range but a lower peak efficiency than Wells turbine. The higher operating range compensates the negative effect of a lower peak efficiency on energy extraction so that the energy extracted by the two turbines is similar.

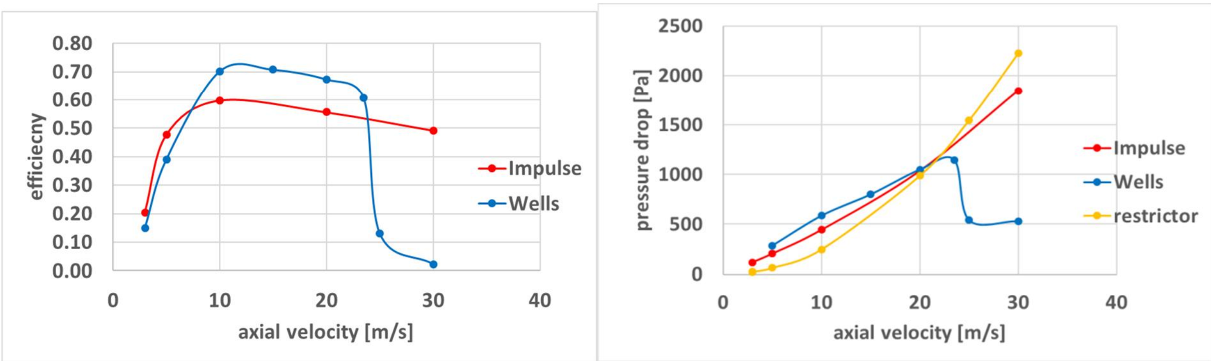

Figure 13 : efficiency and pressure drop of Wells and Impulse turbine as a function of axial velocity

An analysis of the axial loads for the two turbines has also been performed. The results are reported in figure 14. It can be seen how the axial force per blade is much higher for Wells turbine than for Impulse turbine. Moreover, the axial force in the Wells turbine is influenced by the sudden occurrence of stall, and it is strongly dynamic in nature. The higher loads require stronger mechanical structures, whereas a too heavy machine will penalize the inertia of the machine. More severe structural requirements are expected for Wells turbine. The main advantage of the Wells turbine however is its simplicity, since it consists just in a rotor with a limited number of blades. Impulse turbine requires 3 rows of around 40 blades each, with a design for the rotor blades which differs from the design of the stator blades. The drawback of the Wells turbine (especially for low solidity) is the crawling (Raghunatan,1995) which consists in the inability to produce enough torque during startup. In this case, the problem of the turbine speed control during the start-up has to be considered and should be treated in the future.

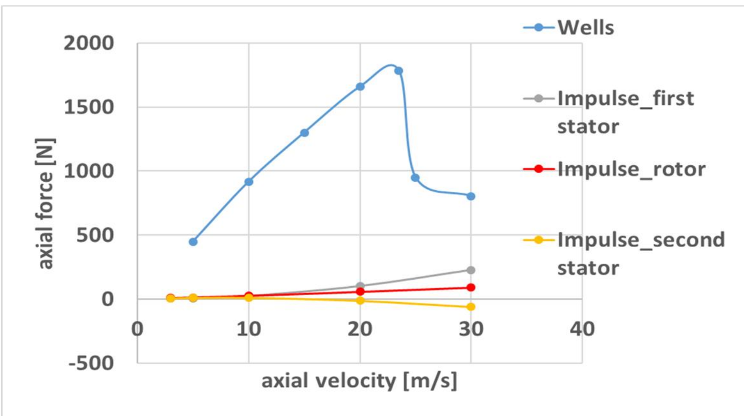

Figure 14 : per blade axial loads for Wells and Impulse turbine

CONCLUSIONS

In this paper it was discussed the possible application of self-rectifying turbines for energy recovery in the Channel Tunnel. By substituting the flow restrictors with self-rectifying turbines a part of the energy dissipated could be recovered. A first design of a Wells turbine and an Impulse turbine was therefore proposed according to 1D analysis and 2D CFD calculations and it was shown that the energy recovery is feasible and in the range between 4.55 and 4.95 GWh depending on the type of turbine and the speed control. To refine these estimations of energy production and to take a decision about the two machines concept, a 3D analysis will be conducted in the next future and experimental tests on small scale models will be then conducted.

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