Cm-scale axial flow water turbines for autonomous flowmeters: an experimental study

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Abstract

This paper reports on the performances and the optimization of a 40 mm diameter water flow energy harvester based on an axial turbine (horizontal axis propeller) coupled to a customized permanent magnet generator. To the best of our knowledge, this work is one of the first comprehensive studies in the low dimensional and flow velocities ranges. The parameters of the propeller have been empirically optimized ($D_l = 1, B = 4$ blades) by the means of experiments in a test-bench pipe at various flow rates from 1 to 9 m$^3$ h$^{-1}$. The best propeller design has been selected for its performances in terms of electrical power and pressure drop. A customized permanent magnet converter with ferromagnetic elements has been modeled, optimized with finite elements simulations and fabricated. This has enabled to increase the electrical power by $2.7 @ 1.5$ m$^3$ h$^{-1}$ (3.2 mW) and up to $14.1 @ 9$ m$^3$ h$^{-1}$ (490 mW) compared to a simple magnet-air coil converter. The figure of merit of our water flow harvester, which takes the electrical power and the pressure drops in the pipe into account, is greater than previously reported water flow energy harvesters, and corresponds to an overall mechanical-to-electrical conversion efficiency of 5.15%.

Keywords: water flow harvesting, axial microturbine, permanent magnet converter

(Some figures may appear in colour only in the online journal)

1. Introduction

Monitoring water distribution networks is becoming increasingly critical not only for utility owners and consumers, but more generally from an environmental point of view. Indeed, according to the French magazine ‘60 Millions of Consumers’ [1], one third of the French cities have a water leakage rate of more than 25% during the distribution. A way to solve this issue consists in monitoring water distribution systems at chosen discrete points in the distribution networks to identify local problems such as corrosion failures or leaks. The information (temperature, pressure, flowrate …) is sent to a base station thanks to a wireless sensor node (WSN) and its RF transmission. With the multiplication of these WSNs, the maintenance costs to change the batteries would inevitably increase, especially when the sensors are located in buried or difficult-to-access areas. A solution, called Energy Harvesting, consists in extracting energy (flows, light, thermal gradients, vibrations,) from the WSN’s environment and to convert it into electricity to power electronic components and functions.

Among ambient energy sources, flow driven energy harvesters can provide a great amount of power, even at
small-scale [2–5]. Indeed, in dusty or dark environments, where solar energy is unreliable or non-existent, and where thermal gradients are insufficient, flow driven harvesters are relevant candidates to power WSNs. In addition, compared to thermal gradient-based energy harvesters, flow driven harvesters intrinsically enable to obtain flow rate measurements since they are inserted into pipes. Using the power of the flow to supply ‘smart flow meters’ appears to be an innovative solution for potable as well as non-potable water networks. Several ducted water flow energy harvesters in the 10–50 cm diameter range have been reported in the state of the art with various types of fluid-to-mechanical conversions. They are overviewed in table 1.

Like water turbines which are mostly found in dams, generating electric power from water kinetic energy is not new. However, centimeter scale water turbines or ‘micro’ turbines have recently become a growing concern to turn pipes into smart systems. Radial-flow micro turbines have been introduced by [6] and were sold by Bosch Junkers [7]. In [10], a cross-flow (or Banki) turbine, coupled to a single phase permanent magnet generator with a claw-pole structure has been proposed. IMTEK [5, 6] also reported a radial-flow turbine incorporating a two-pole ring magnet and three induction coils connected in a star circuit. The drawbacks of this kind of fluid-to-mechanical conversion, i.e. vertical axis turbines, is (i) the pressure drop which is often significant (a few bars) compared to the input pressure and (ii) the tightness issues to take into account. As water in a city is distributed by extensive piping networks, keeping the pressure drop low across pipes is a very important task to avoid excessive pumping power requirements. In 2016, CEA-Leti [4] proposed a centimeter-scale axial flow turbine converting the rotation of a horizontal-axis propeller into electricity thanks to distributed magnets of alternate polarities at the rotor periphery and air coils outside the pipe. This harvester induced low pressure drops (<0.05 bars) compared to the state of the art and delivered more than 400 μW @ 41 min−1 which is enough to supply flow rate and temperature sensors with a low duty cycle. In 2017, Adamski et al. [8] published the same concept in additive manufacturing process but with a reduced size. To the best of our knowledge, none of the above-mentioned references studied the influence of the propeller design on the device efficiency. Furthermore, the literature reveals a lack of measurements concerning the pressure drop generated by the harvester in the pipe. This work follows in the footsteps of [4] by proposing a detailed experimental study and the optimization of an axial-flow micro turbine in a DN40 pipe.

Section 2 introduces the operating conditions of the turbine and its dimensional parameters. In section 3, details of the turbine design are presented and the experimental performances of various types of propellers are reported. The electromechanical converter is also detailed and optimized thanks to finite elements methods (FEM). Test bench conditions and measurement results are given in section 4. Finally, the propeller’s power coefficients and maximum end-to-end efficiencies are computed to compare our harvester to the state of the art.

2. Dimensions and flow regimes

Circular pipes with a diameter of 40 mm have been selected as a starting point for this study. This value has been chosen to limit the impact of the side effects (periphery of the pipe and hub), while remaining in a low-dimensional range. Furthermore, our harvester was designed primarily for deployment in water pipes where a flow rate $Q$ in the range $[1–10 \text{ m}^3 \text{ h}^{-1}]$ can be expected.

Contrary to free stream conditions, when the water flow circulates within a closed conduit, its velocity changes from zero at the surface to a maximum at the pipe center which has a significant impact on the propeller’s efficiency. The velocity profiles in laminar and turbulent regimes in fully developed flow are illustrated in figures 1(a) and (b) respectively.

The velocity profile of the flow depends on the ratio of inertial forces to viscous forces in the fluid. This ratio is the Reynolds number based on the pipe diameter $R_{ep}$, expressed for internal flow in a circular pipe as (1):

$$R_{ep} = \frac{\rho v c D}{\mu} = \rho Q / 2 \mu \pi R,$$

where $\tau$ is the average flow velocity ($\text{m s}^{-1}$), $D$ the diameter of the pipe (m), $\rho$ the density of the fluid ($\text{kg m}^{-3}$) and $\mu$ the dynamic viscosity of the fluid (Pa s). Considering that $\tau = Q / \pi R^2$ with $Q$ varying between 1 and $10 \text{ m}^3 \text{ h}^{-1}$, $R_{ep}$ always exceeds 10 000 which shows that the flow regime is mainly turbulent in these conditions. The velocity profiles of the flow, of which knowledge is required to work out the coefficients of kinetic energy for example, can be determined with analytical methods for laminar regimes (Poiseuille laws). Nevertheless, this is not the case for turbulent regimes for which numerical methods and turbulence modeling have to be used. This was outside the scope of the current project. Then, the average velocity of the water $\bar{v}$ related to the flow rate (expressed by $\bar{v} = Q / \pi R^2$) is used for the input specifications. With these operating conditions, the flow velocity ranges between 0, 2 m s$^{-1}$ (1 m$^3$ h$^{-1}$) and 2 m s$^{-1}$ (10 m$^3$ h$^{-1}$).

The flow regime seen by the blade of the propeller must also be evaluated because it affects the blade performance and, consequently, impacts the blade design. Then, the Reynolds number based on the blade chord length, $Re_e = \rho v c / \mu$, with $v$ the blade velocity at the tip is used. Over the range of the rotation frequencies reachable for such conditions (i.e. down to 5 Hz and up to 30 Hz) and with $c = 10 \text{ mm}$, $Re_e$ can be as low as 2000 and never exceeds 30 000. At such low Reynolds numbers, the negative effects of viscous losses are amplified and blades operating in this regime exhibit performances worse than the ones of blades in a fully turbulent regime [2, 11]. Since Navier–Stokes equations are nonlinear, it is very complicated to build analytical models of the propellers, especially if interactions between the rotor and the duct are considered. Furthermore, even if some software using computational fluid dynamics exist, the Multiphysics couplings between the fluid, the propeller, the duct and the electromechanical converter can be a very hard task to model and real experiments must be conducted to validate them. Consequently, empirical methods have been used to analyze the behavior and the performances of various propellers in a ducted water pipe.
This work was designed and built to harvest energy from a horizontal axis, lift-based water flow energy harvester. The schematic of the overall device is shown in figure 2, with the permission of AIP Publishing. https://doi.org/10.1063/1.5040712.

3. Design and fabrication

3.1. Water-flow energy harvester concept

The energy harvester exploits a horizontal axis, lift-based propeller for the fluid-to-mechanical conversion and a magnet-coil architecture for the mechanical-to-electrical conversion. The rotor is composed of a shaft and a horizontal axis propeller with magnets of alternate polarities distributed at its periphery. For the stator, a casing machined in nonmagnetic material (Polyoxymethylene) embodies nine air coils located outside the pipe which surround the rotor at its periphery. A schematic of the overall device is shown in figure 2(a) and a picture of the final prototype is shown in figure 2(b). The harvester alone (i.e. without sleeves) is 45 mm wide; its external diameter is 85 mm.

As soon as water flows in the harvester, the propeller starts rotating, leading to a rotational movement of the propeller and thus of the magnets. The movement of the magnets induces a variation of magnetic flux in the coils, which is finally turned into electricity thanks to Lenz’s law. Further details on the main parts are given in the following sections.

3.2. Propeller design

3.2.1. Schmitz theory. The propeller’s design is often defined by three main parameters which are (i) the pitch angle, (ii) the chord length and (iii) the blade profile. Indeed, it has been proven that much can be used from the design and operation of wind turbines [12] and ship propellers [13]. Schmitz’s theory [14], borrowed from the dimensioning of wind turbines, has been used to optimize the chord and the angle of the blades. Compared to Betz’s law [15], this design takes the downstream rotation of the wake (in opposite direction to the rotor) into account. To extract the maximum power, and according to Schmitz’s theory, the pitch angle $\beta(r)$ and the chord length $c(r)$ should be:

$$\beta(r) = \frac{2}{3} \arctan \left( \frac{R}{r \lambda_D} \right) - \alpha_D$$

where $\lambda_D$ is the tip speed ratio, $R$ is the propeller radius, and $c(r)$ is the blade chord.

### Table 1. Overview of state-of-the-art cm-scale water flow energy harvesters.

| References | Figures | Pipe diameter | Type | Maximum pressure drop | Output power |
|------------|---------|---------------|------|------------------------|--------------|
| This work  | ![image](https://doi.org/10.1063/1.5040712) | 41, 5 mm (DN40) | Horizontal axis propeller, coil-magnet converter | <0.06 bars @ 150 l min\(^{-1}\) | 550 mW @ 150 l min\(^{-1}\) |
| [6] Becker et al 2013 | ![image](https://doi.org/10.1063/1.5040712) | ~15 mm | Vertical axis propeller, Induction coils + two-pole ring-magnet | N.A | 300 mW @ 191 l min\(^{-1}\) 30 mW @ 51 l min\(^{-1}\) |
| [5] Hoffmann et al 2013 | ![image](https://doi.org/10.1063/1.5040712) | ~15 mm | Vertical axis propeller, Induction coils + two-pole ring-magnet | 2, 2 bars @ 201 l min\(^{-1}\) | 720 mW @ 201 l min\(^{-1}\) 2 mW @ 31 l min\(^{-1}\) |
| [7] Commercial (Bosch Junkers) | ![image](https://doi.org/10.1063/1.5040712) | ~10 mm (nozzle) | Vertical axis propeller (cross flow), permanent magnet generator | N.A | 265 mW @ 51 l min\(^{-1}\) |
| [4] Boisseau et al 2016 | ![image](https://doi.org/10.1063/1.5040712) | 20 mm | Horizontal axis propeller, air coil-magnet converter | <0.05 bars @ 201 l min\(^{-1}\) | 5.76 mW @ 101 l min\(^{-1}\) 25 mW @ 201 l min\(^{-1}\) |
| [8] Adamski et al 2017 | ![image](https://doi.org/10.1063/1.5040712) | 12, 7 mm | Horizontal axis propeller + coil-magnet converter | N.A | 4 mW @ 141 l min\(^{-1}\) 0.5 mW @ 7.5 l min\(^{-1}\) |
| [9] Alrowaijeh et al 2018 | ![image](https://doi.org/10.1063/1.5040712) | 12, 7 mm | Vertical axis propeller | N.A | 2.1 mW @ 13.3 l min\(^{-1}\) |
| [10] Zenerino et al 2012 | ![image](https://doi.org/10.1063/1.5040712) | ~10 mm | Vertical axis propeller + claw pole generator | N.A | 60 mW @ 51 l min\(^{-1}\) |

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(2) Reproduced from [10], under CC BY 3.0 license. https://doi.org/10.5772/50719.
With $\lambda_D$ the design tip speed ratio (TSR) $\lambda_D = \omega R / \nu_{\text{fluid}}$, $\alpha_D$ the angle of attack chosen to maximize the lift-to-drag ratio, $B$ the number of blades and $C_L$ the lift coefficient which depends on the chosen profile, the Reynolds number (see further below) and the angle of attack $\alpha_D$. The pipe and the blade cross-sections are illustrated in figure 3, showing the main parameters of the propeller.

3.2.2. Parameters

3.2.2.1. Choice of the angle of attack $\alpha_D$ and lift coefficient $C_L$. At these low Reynolds numbers, it is predicted that thin, rough and slightly cambered (2%–4%) profiles enable to reach lift-to-drag ratio from 1 to 15 [11]. In order to avoid the blade to stall (separation occurs, so lift coefficient would decrease), the angle of attack $\alpha_D$ of all blades has been set to 6°, for which the value of the lift coefficient $C_L$ should be comprised between 0, 5 and 1 [16].

3.2.2.2. Choice of the number of blades. In linear momentum theory, the greater $B$, the higher the efficiency (tip losses are reduced) but it leads to use lower chord lengths which negatively affects the Reynolds number and thus decreases the lift-to-drag ratio of the blade. It should also be emphasized that in our case, the rotor is composed of a magnet ring which surrounds the blades at their tips. Hence, in practice, the effect of the number of blades can differ from theoretical assumptions. Thus, we propose to empirically study the effect of the blade number on the turbine performance by designing two group of propellers: the first group with $B = 4$ blades and the second one with $B = 6$ blades.

3.2.2.3. Choice of the ‘design TSR’ $\lambda_D$. When using linear momentum theory to analyze ideal horizontal axis turbines with wake rotation, it can be easily shown that the higher the design TSR, the greater the maximum theoretical efficiency. Practically, the efficiency of the propeller decreases when the blades turn at a different TSR than the one for which they have been designed. Drag and tip losses that are functions of the total number of blades reduce the power coefficients of turbines. We propose to empirically study the effect of $\lambda_D$ on the turbine performance by analyzing the results for 3 values of $\lambda_D$: 0.5, 1 and 2.

Thus, six propellers with two different number of blades ($B = 4$ and 6) and three different ‘design TSRs’ ($\lambda_D = 0.5$, 1 and 2) have been fabricated. The pitch angle $\beta(r)$ and the

$$c(r) = \frac{16\pi R}{BC_L} \sin^2 \left( \arctan \left( \frac{R}{3\lambda_D} \right) \right).$$

Figure 1. Velocity profiles of a ducted flow in (a) laminar and (b) turbulent regime.

Figure 2. (a) Exploded schematic and (b) assembled harvester.
and the angular UN d resistances are around 750 Ω. To convert mechanical 3.4.1. General configuration

3.4. Electromechanical converter

(3.4.2. FEM simulations of the electromechanical converter.

3.3. Fabrication of the propellers

The six propellers have been fabricated using a 3D-printing process (Laser sintering) in polyamide powder filled with glass particles. This material provides higher thermal resistance (up to 110 °C) than classical polyamides, enabling to use the harvesters in various types of applications, including district heating and cooling systems. FEM simulations have been used to check the mechanical resistance of the propellers to high flow rates (up to 10 m³·h⁻¹). A picture of the six propellers is given in figure 5.

3.4. Electromechanical converter

3.4.1. General configuration. To convert mechanical rotations into electricity, the electromagnetic conversion appears to be very relevant in terms of power density compared to the electrostatic conversion [17] and reliability (including resistance to ageing) compared to the piezoelectric conversion [18]. A customized permanent magnet generator has been designed to convert the mechanical rotation of the propeller into electricity. The rotor is composed of several magnets of alternate polarities distributed at the periphery of the propeller. The stator is made of several coils located outside the pipe and facing the magnets. As shown in figure 6, the number of coils and magnets has been optimized to simplify the power management circuit since three groups of coils generate the same voltage waveforms simultaneously, and can be connected in series.

Air coils with an external diameter of 10 mm, a height of 10 mm and a wire cross section of 45 μm have been used. The number of turns of each coil is 2800 and their winding resistances are around 750 Ω. For reasons of space requirement and mechanical robustness of the pipe, the number of coils has been set to nine. Square magnets (NdFeB, N52 type) of 5 × 10 × 10 mm have been used; these dimensions offer a good mechanical robustness of the ring while preserving a strong magnetic field through the coils.

Since the rotating magnets should produce the strongest gradient of flux (dΦ/dθ) possible, we propose to use CHC screws made of steel, inserted inside the coils. As shown in figure 7, these ‘magnetic cores’ fulfill two primary functions, namely, (i) confining, guiding and amplifying the magnetic flux through the coils and (ii) ensuring the mechanical holding of the coils in the tube. As explained further below, this solution offers a low cost solution to considerably increase the output power of the harvester.

Thanks to FEM simulations, the minimum distance of 1.15 mm between the tip of the screw and the inside periphery of the pipe has been set to withstand 10 bars of static pressure at 90 °C.

3.4.2. FEM simulations of the electromechanical converter. Faraday’s law of induction gives an expression of the voltage across the coil induced by the magnets motion (4):

\[ U_{\text{open}} = -N \frac{d\Phi}{dt} \]  

(4)

with \( N \) the coil’s number of turns and \( \Phi \) the magnetic flux (Wb) captured by the coil surface, which is the integral of the normal component of the magnetic field \( B \) passing through that surface. The magnet angular velocity \( \omega \) and the angular variation of the magnetic flux captured by the coil surface can be made independent from each other thanks to equation (5):

\[ U_{\text{open}} = -N \frac{d\Phi}{d\theta} \frac{d\theta}{dT} = -\beta \omega \]  

(5)

with \( \beta \) the generator coefficient in voltage per unit rotation speed \( (\beta = N \frac{d\Phi}{d\theta}) \) and \( U_{\text{open}} \) the voltage across the coil in open circuit.

FEM simulations with FLUX 3D (Altair) have been performed to observe the voltage across the coils at constant rotation frequency. Figure 8(a) depicts a 3D transient magnetic modeling of the generator with 24 magnets and with magnetic cores inside the air coils. Using these simulations, the \( \beta \) coefficients were computed for two converters configurations (with and without magnetic cores) and various number of magnets around the propeller (figure 8(b)).

As expected, the simulations results presented in figure 8 suggest that the higher number of magnets, the higher \( \beta \) coefficient becomes. We can also note that the addition of magnetic cores (screws) increases the output voltage by up to 4.6. This means that the output power can be improved by a factor of 21 at imposed rotation speeds thanks to the magnetic cores.
4. Experimental results

4.1. Test bench and experimental method

Experiments were carried out at CEMITEC in a 200 cm long DN40 pipe. Water flows were measured with a Burkert 8041 flow meter and pressure drops with a Deltabar PMD75 (Endress+Hauser Ltd) differential pressure transmitter with piezo-resistive sensors. The pressure sensors are positioned at about 100 cm downstream and upstream of the harvester. A picture of the test set-up is shown in figure 9(a) with a focus on the harvester in figure 9(b).

The six proposed propellers designs have been tested at three different flow rates (1, 3, 6 m$^3$ h$^{-1}$). The experiments were performed by varying identical electric loads connected to each coil after setting the water pipe to a fixed flow rate. The rms voltage, the rotation frequency, the pressure drop and the flow rate have been measured for twenty resistive loads, from short-circuit to 10 MΩ.

4.2. Measured performances without magnetic cores

The purpose of these tests was to empirically find the best propeller in the specified flow range. Since there is a strong magnetic interaction between the permanent magnets of the rotor and the screws, it has been decided not to install the screws on the harvester in a first time. The first tests consisted in measuring the performance of the six different designs at various flow rates. The total electrical power generated by the harvester is given by (6):

$$ P_{\text{elec}} = 9V_{\text{rms}}^2/R_{\text{load}} $$

With $V_{\text{rms}}$ the root mean square voltage across $R_{\text{load}}$, the nine coils showing the same voltage waveforms and root mean square values.

The measurement results are given in figures 10(a)–(c) where the electrical powers are plotted as a function of the operational TSR $\lambda_o$ defined by (7):

$$ \lambda_o = \frac{2\pi f_{\text{rot}} R}{\nu} $$

with $f_{\text{rot}}$ the measured rotation frequency of the propeller.

As expected, those measurements show that the propellers which operate at highest $\lambda_o$ output the highest electrical powers. Five designs (Nb4_L05, Nb4_L1, Nb4_L2, Nb6_L05 and Nb6_L1) exhibit similar behaviors: their operational lambda are relatively constant with the flow rate and are not very affected by the variation of resistive loads, even at low flow rates. This dependency with the loads decreases even more as the flow rate increases. The performances of Nb4_L2 and Nb6_L2 (especially) are poor in terms of output power. This is probably due to the fact that the blade’s chord is very small compared with the other designs, which negatively affects the chord Reynolds number and thus the aerodynamic performance of the blade (lift-to-drag ratio decreases). One can note that Nb4_L2’s behavior is very dependent on the resistive load and on the flow rate variations. As its performances improve with the increase in flow rate, it can be guessed that this design is best suited to higher flow velocities.

To sum up, the best performances at low flow rates are reached with designs for which $\lambda_o = 1$ where 370 μW @ 1 m$^3$ h$^{-1}$ are harvested for both Nb4_L1 and Nb6_L1. At high flow rates, the designs for which $\lambda_o = 1$ are also the most powerful propeller designs: 13 mW and 14 mW @ 6 m$^3$ h$^{-1}$ for Nb4_L1 and Nb6_L1 are generated respectively. The design Nb4_L2 improves as the flow rate increases (14 mW @ 6 m$^3$ h$^{-1}$) but one can notice that its electrical output power
is more dependent on the electrical loading than Nb6_L1 and Nb4_L1. This means that the total electrical power is not negligible compared with the mechanical power. A summary of the turbines’ performances is depicted in figure 11(a), where the maximum electrical powers are plotted as a function of the disk area ratios for which a particular \( d_l \) and a number of blades \( B \) can be associated. Figure 11(b) shows the pressure drops of the various propeller designs as a function of their disk area ratios for various flow rates.

It can be noticed that the propeller designs with \( d_l = 0.5 \) show higher pressure drops at low flow rates and lower pressure drop at high flow rates compared with the other designs. The propeller designs for which \( d_l = 1 \) and 2 have a similar behaviors in terms of pressure drop (\(<1 \text{mbar} @ 1 \text{m}^3 \text{h}^{-1}, \approx 5 \text{mbar} @ 3 \text{m}^3 \text{h}^{-1} \) and \( \approx 20 \text{mbar} @ 6 \text{m}^3 \text{h}^{-1} \)). For the following experiments, design Nb4_L1 (\( B = 4, \lambda_d = 1 \)) has been selected for its best performances in terms of electrical power and pressure drop in the whole flow rate range.

### 4.3. Measured performances with magnetic core: final harvester

Figure 12 depicts the measurement results of the final propeller design (Nb4_L1; \( B = 4, \lambda_d = 1 \)) with 24 magnets at the periphery of the rotor and with the magnetic cores inserted.
into the coils for the stator, at flow rates of 1.5, 3, 6 and 9 m$^3$ h$^{-1}$. The lowest flow rate (1.5 m$^3$ h$^{-1}$) comes from the fact the harvester does not start to rotate below this value. This is because the cogging torque increases the cut-in speed of the harvester (>1.5 m$^3$ h$^{-1}$, i.e. 0.31 m s$^{-1}$) while it is around 0.5 m$^3$ h$^{-1}$ (0.1 m s$^{-1}$) without magnetic cores. Nevertheless, these figures clearly show the benefit of the ‘magnetic cores’: the maximum electrical powers increase a lot when screws are settled on the stator: $P_{elec} = 3$, 2 mW @ 1, 5 m$^3$ h$^{-1}$ (i.e. +270%); 41 mW @ 3 m$^3$ h$^{-1}$ (i.e. +1030%); 220 mW @ 6 m$^3$ h$^{-1}$ (i.e. +1390%); 490 mW @ 9 m$^3$ h$^{-1}$ (i.e. +1410%).

These results also show that:
- Due to the magnetic losses, the ‘open-circuit’ (high $R_{load}$) rotation frequency of the turbine is lower when the magnetic cores are set inside the air coils and especially at low flow rates.
- The rotation frequency of the harvester is much more impacted by the resistive load for the ‘magnetic cores’ versions, especially at low flow rates.

The magnetic cores increase the electromechanical coupling of the harvester. They have two benefits: (i) by increasing the magnetic flux variation through the coils, it
increases the electrical output power at constant rotation speed and (ii) it enables to slow the propeller towards a more optimized rotation speed, where the fluid-to-rotation efficiency is higher and therefore where the shaft power is higher. As expected, the harvester with magnetic cores generates more pressure drops than the one without cores, namely, 2.6 mbar (+60%), 9.5 mbar (+40%), 30.2 mbar (+14%) and 64 mbar (+14%) for 1.5 m$^3$ h$^{-1}$, 3 m$^3$ h$^{-1}$, 6 m$^3$ h$^{-1}$ and 9 m$^3$ h$^{-1}$ respectively. Nevertheless, these pressure drops remain within meaningful values compared with the pipe input pressure (2 bars in the test bench).

4.4. Efficiency of the harvester

When applying the conservation of energy principle in an horizontal pipe, pressures and velocities are ruled by the equation of Bernoulli (8):

$$\frac{p_1}{Q} + \frac{1}{2} \rho v_1^2 + \frac{P_{\text{tot}}}{Q} = \frac{p_2}{Q} + \frac{1}{2} \rho v_2^2$$

(8)

with $Q$ the flow rate (m$^3$ s$^{-1}$), $P_{\text{tot}}$ the total power extracted by the turbine (W), $p_1$ and $p_2$ respectively the downstream and upstream pressures of the fluid (Pa) and $v_1$ and $v_2$ the downstream and upstream fluid velocity (m s$^{-1}$).

Assuming that $v_1 = v_2$ in a pipe of constant section, equation (8) can be simplified in (9):

$$P_{\text{tot}} = Q(p_2 - p_1) = Q\Delta p$$

(9)

with $\Delta p$ the pressure drop generated by the harvester in the pipe.

The mechanical input power $P_{\text{mech}}$ provided by the propeller to the shaft can be approximated by equation (10)

$$P_{\text{mech}} = T \cdot \omega \approx P_{\text{elec_tot}} + P_{\text{bearing_loss}} + P_{\text{Core_loss}}$$

(10)

with $T$ and $\omega$ the driving torque and angular speed of the propeller respectively, $P_{\text{elec_tot}} = P_{\text{elec_loss}} + P_{\text{elec_harv}}$ the electrical power converted into $R_{\text{int}}$ (resistive losses) and $R_{\text{load}}$ respectively, $P_{\text{bearing_loss}}$ the mechanical losses wasted in the bearings and $P_{\text{Core_loss}}$ the total iron losses inherent to the magnetic cores (hysteresis and Eddy currents losses).

Then, the aerodynamic performances of the propeller are measured by calculating the power coefficient $C_p$ defined by equation (11)

$$C_p = \frac{P_{\text{mech}}}{Q\Delta p} = \frac{T \cdot \omega}{Q\Delta p}.$$  

(11)

And the end-to-end efficiency of the harvester $\eta_{\text{tot}}$ can be defined as (12)

$$\eta_{\text{tot}} = \frac{P_{\text{elec_harv}}}{Q\Delta p}.$$  

(12)
Figure 11. (a) Maximum electrical powers and (b) pressure drop generated by the six propellers as a function of their disk area ratios for various flow rates.

Figure 12. Electrical power of ‘Nb4_L1’ design (24 magnets) with and without magnetic cores for various flow rates (a) 1.5 m$^3$ h$^{-1}$, (b) 3 m$^3$ h$^{-1}$, (c) 6 m$^3$ h$^{-1}$ and (d) 9 m$^3$ h$^{-1}$.
In our application, measuring the propeller’s torque is complicated for two reasons: (i) the environment is immersed and pressurized and (ii) the size of the potential torque sensor compared to the dimension of the harvester may modify the aerodynamic performances of the propeller. Accordingly, an indirect measurement of $P_{\text{mech}}$ has been performed on the device by measuring $P_{\text{elec...loss}}$, $P_{\text{bearing...loss}}$ and $P_{\text{Core...loss}}$.

We measured the bearing losses, thanks to a ‘let it roll’ experiment. This experiment consists in quantifying the frictional torque by measuring the decrease in the angular speed $\omega$ of the rotor as a function of the time after giving it an initial spin. Assuming that both dry friction and fluid drag are simultaneously acting on the rotor, the angular speed is given by (13):

$$\omega(t) = (\omega_0 + a/b)e^{-bt} - a/b.$$  

(13)

Nonlinear fits of the angular speed for various ‘let it roll’ experiments [19] enabled to get the mean dry friction coefficient ($a \approx 50, 2 \times 10^{-6} \text{N m}$) and the drag friction coefficient ($b \approx 41 \times 10^{-8} \text{N m s rad}^{-1}$) for the considered propeller. Using these two coefficients, the power losses in the bearings can be calculated by equation (14):

$$P_{\text{bearing...loss}} = \tau \cdot \omega = (a + b\omega)\omega$$  

(14)

with $\tau$ the resistive torque due to the plain bearings.

Since core losses are not known for the chosen material, we estimated the core losses $P_{\text{Core...loss}}$ of the harvester’s magnetic component (an air coil and its magnetic core). For those characterizations, we build a test bench which imposes the same magnetic flux as the measurements in the pipe. This was carried out by including the magnetic component inside a large excitation coil across which a sinusoidal voltage was applied by a power amplifier and a signal generator. The core losses were then estimated by measuring the difference between the reactive powers delivered to the excitation coil with and without the magnetic component inside it and at equal input rms current.

Figures 13(a) and (b) show the power coefficient $C_p$ and the end-to-end efficiency $\eta_{\text{tot}}$ of the harvester as a function of the operational lambda $\lambda_o$, for various flow rates.

Figure 13(a) shows that the best propeller exhibits a maximum coefficient of power of $C_p = 14\%$ at a $3 \text{ m}^3\text{ h}^{-1}$ flow rate. One can note that this maximum value could easily be outperformed by slowing the propeller down, more than it actually is. This can be obtained by increasing the electromagnetic coupling of the turbine. For that purpose, a solution would be the increase of the magnets and coils dimensions to improve the magnetic flux variation captured by the coil surfaces.

4.5. Comparison to the state of the art

To compare our harvester with other state-of-the-art devices, the maximum end-to-end efficiencies $\eta_{\text{tot...max}}$ (i.e. with maximum $P_{\text{elec...loss}}$) are plotted as a function of the flow rate in figure 14. It shows that, despite the increase in pressure drops, the ‘magnetic cores’ version of our harvester shows better performances than the harvester without magnetic cores. Despite a lack of data concerning the pressure drop of the state-of-the-art harvesters, a comparison was made with two harvesters [4, 5]. Our harvester has good performances at high flow rates but mixed results in the low flow rates range. Best performances are expected for [5] in the low flow rate range ($<1 \text{ m}^3\text{ h}^{-1}$), but the induced pressure drops are a hundred time higher than those generated by our harvester ($5 \text{ mbar} \ @ \ 201 \text{ min}^{-1}$ for our harvester versus 2.2 bar @ $201 \text{ min}^{-1}$ for [5]). Above $1 \text{ m}^3\text{ h}^{-1}$, the efficiency of our device outperforms the state of the art in these dimensional and flow rate ranges, reaching an end-to-end efficiency up to 5.15% at $3 \text{ m}^3\text{ h}^{-1}$.

5. Conclusion

Our work is the first detailed experimental study of a centimeter-scale water flow energy harvester. It has enabled to optimize and characterize a water flow energy harvester based on a horizontal axis propeller coupled to a customized permanent magnet generator. Compared to cross-flow devices, horizontal axis propellers enable to harvest electrical power without generating high pressure drop in the pipe and thus
avoiding excessive pumping power requirements. Thanks to a comparative study of six propellers with different design parameters, a 4-blade propeller having a design TSR of 1, an angle of attack of $6^\circ$ and a NACA2515 profile has been selected. The design has been selected for its best performances in terms of electrical power (from 0.370 mW @ $1 \text{ m}^3 \text{ h}^{-1}$ to 14 mW @ $6 \text{ m}^3 \text{ h}^{-1}$) and its low pressure drops (18, 6 mbar @ $6 \text{ m}^3 \text{ h}^{-1}$). The permanent magnet generator of the harvester has also been modeled and optimized, which has led to an increase of the electrical power by up to 270% @ 1, 5 m$^3$ h$^{-1}$ (3, 2 mW) and 1410% @ 9 m$^3$ h$^{-1}$ (490 mW) with the same propeller. Our harvester, inserted in a DN40 pipe, has a cut-in speed of 1.5 m$^3$ h$^{-1}$ (251 min$^{-1}$) and generates less than 3 mbar at 1, 5 m$^3$ h$^{-1}$ and 64 mbar @ 9 m$^3$ h$^{-1}$ of pressure drops which is far less than the state of the art. The maximum end-to-end efficiencies $\eta_{\text{tot, max}}$, which takes into account the electrical power and the pressure drop, is greater than previously-reported water flow energy harvesters (5.15% @ 3 m$^3$ h$^{-1}$). Many improvements can be implemented to increase the performances of the harvester. For example, the permanent magnet generator can be further optimized by slowing down the rotation of the propeller even more, to reach the optimal operating point of the propeller (i.e. a maximum mechanical input power). This must not be performed at the expense of the pressure drops and the cut-in speeds. This experimental study, along with the proposed water flow harvester, paves the way towards the deployment of wireless and battery-less autonomous flow meters for various application domains such as agriculture (irrigation systems), housings or districts heating and cooling systems.

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