Design of a hydroformed metal blade for vertical-axis wind turbines

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Vertical-axis wind turbines (VAWTs) have experienced a renewed impulse during the last few years, with important research efforts focused on them. This work explores whether the global profitability of VAWTs can be improved through improved manufacturing techniques. We studied how large-series production techniques from the sheet-metal industry can be used to create blades of H-type Darrieus turbines. Blade size and shape were determined via aerodynamic and structural analyses. The proposed solution is based on the use of hydroforming manufacturing techniques with metal sheets. Our estimations show that with the positive effects of a large-scale use and production (economies of scale), such metal blades have a 90% reduction potential in their production costs compared to fibre-reinforced ones for single turbines.

Keywords: Vertical-axis wind turbines, blade manufacturing, cost analysis

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I. INTRODUCTION

Horizontal axis wind turbines (HAWTs) have been the focus of most of the wind energy related research during the last few decades and represent the major portion of the installed capacity. However, research work on vertical-axis wind turbines (VAWTs) continued in parallel, usually focused in the small scale, where different configurations and approaches have been proposed. It has been stated that VAWTs are suitable for electricity generation in the conditions where traditional HAWTs are unable to give reasonable efficiencies, such as turbulent winds with strong variations of the wind direction. Another important advantage is the fact that VAWTs are omnidirectional, accepting wind from any direction without any yawing mechanism\(^1\). Recent research has also shown that it is possible to increase the global performance of VAWTs by erecting them in tight arrays with alternating directions of rotation. A column of co-rotating VAWTs was estimated to provide a power increase of 50\% to 100\%, compared to the same number of isolated VAWTs\(^2\), confirming the validity of the concept previously suggested by Dabiri\(^3\). In this manner, farms of VAWTs can achieve a higher power output per unit land area (W/m\(^2\)) and consequently a smaller footprint than equivalent HAWT farms. Other options to increase the power output of VAWTs, such as inlet guide-vanes\(^4,5\) have also been explored.

Yet despite all the recent advances, VAWTs are economically not yet competitive against HAWTs\(^6\), due to their lower power coefficients. In this work we therefore focus on ways to improve their global profitability, by investigating the use of low cost production technologies together with the choice of the materials used for blade production. The blades of H-type VAWTs, also known as giromills, are particularly suited to low-cost production, because of their simple shape (no taper, no twist).

For a new blade with equal aerodynamic performance compared to existing blades, decreasing the manufacturing cost is a straightforward way to improve the return on investment. A low manufacturing cost is typically achieved by using large-series production technologies. It has recently been suggested that large farms of closely-spaced VAWTs could achieve better power densities than the classic existing HAWT farms\(^2,3\). The market uptake of this insight would require the use of very large volumes of turbines (and blades), thereby increasing the need for large-series production technologies.

Of course, the choice of a manufacturing process that is suited for large-series production
strONGLY DEPENDS ON THE MATERIAL TO BE USED. MOST WIND TURBINE BLADES ARE CURRENTLY MADE OF FIBRE-REINFORCED COMPOSITES, WHICH ARE QUITE COSTLY (UP TO 30% OF THE WHOLE TURBINE COST). METAL IS SOMETIMES USED AS AN INTERNAL STRUCTURAL COMPONENT IN THE BLADES OF LARGE HAWTS; SOME SMALL WIND TURBINES HAVE EXTRUDED ALUMINIUM BLADES. WE ARGUE THAT THE USE OF METAL COULD FACILITATE THE PRODUCTION PROCESS OF THE TURBINE BLADES, AND WOULD ADDITIONALLY HAVE ADVANTAGES WITH RESPECT TO ISSUES LIKE ROBUSTNESS, SUSTAINABILITY AND RECYCLING. THE WORKFLOW LEADING TO SUCH A BLADE IS REPRESENTED IN FIG. 1. IF METAL BLADES ARE TO GAIN A LARGER SHARE OF THE MARKET, THEY SHOULD LEAD TO A DECREASE IN THE COST OF ENERGY BY OFFERING (AT LEAST) EQUAL AERODYNAMIC PERFORMANCE, DECREASED COST (MATERIALS, MANUFACTURING PROCESS) AND INCREASED SUSTAINABILITY (ENERGY FOR PRODUCTION, RECYCLING, LIFETIME).

THE AERODYNAMIC PERFORMANCE OF THE BLADE IS DETERMINED BY ITS EXTERNAL SHAPE AND, TO A LESSER DEGREE, ITS SURFACE ROUGHNESS. IN SECT. II WE DISCUSS THE OVERALL DIMENSIONING OF THE ROTOR AND THE AERODYNAMIC DESIGN OF THE BLADES, INCLUDING THE PROFILE CHOICE. IN SECT. III WE INVESTIGATE THE STRUCTURAL ASPECTS OF THE METAL BLADE, IN PARTICULAR DEFORMATION, STRESS AND FATIGUE. IN SECT. IV WE PROPOSE SUITABLE PRODUCTION TECHNIQUES SO THAT, AT SUFFICIENTLY LARGE PRODUCTION VOLUMES, METAL BLADES ARE SIGNIFICANTLY CHEAPER THAN COMPOSITE BLADES.

FIG. 1. WORKFLOW FOLLOWED FOR THE ANALYSIS AND DEVELOPMENT OF THE LOW-COST BLADES.

II. AERODYNAMIC ANALYSIS OF VAWT BLADES

A. Selected method

The aerodynamics of a VAWT is quite different from a HAWT (see e.g. Manwell, McGowan, and Rogers) and in general more complicated. The relative motion of the blades
with respect to the freestream velocity can cause dynamic stall, particularly at low tip-speed ratios. The struts have a significant influence both on the performance of the turbine and the pattern of the resulting flow. Tip-vortices further add to the complexity of the flow pattern.

As a result, the flow field is fully three-dimensional with complex flow structures, and it represents a challenging aerodynamic problem. Many authors have made predictions of turbine performance by means of Computational Fluid Dynamics (CFD), with mixed results. When 2D simulations are conducted the results usually overestimate by about a 100% the power coefficient of the turbine. Higher accuracy can be achieved with 3D models and/or Large Eddy Simulations (LES) turbulence models, but the observed errors were still over 30% for a wide range of tip-speed ratios. These methods are computationally much more expensive, making them hard to integrate into the design process.

Apart from CFD, the numerical simulations of VAWT aerodynamics is often done using either vortex models or stream tube models. A nice overview of different methods to study Darrieus turbines is given in Jin et al. Vortex models typically achieve good accuracy but still require a considerable computational effort. The most advanced stream tube model is the Double Multiple Stream Tube (DMST) model. Such models are relatively easy to calculate, while still offering a reasonably good prediction of the turbine’s performance. Therefore, they have been widely used and are adopted here for the overall performance estimates, while CFD is used for 2D and 3D blade aerodynamics calculations.

### B. Design parameters of the rotor

The dimensioning of the rotor is the first choice to be made when designing a wind turbine. The rotor size determines the maximum attainable power and is one of the main factors contributing to the investment cost of the turbine. The rotor size of the prototype is chosen to be representative of existing VAWTs while small enough to keep the research and development costs under control.

The maximum blade length will also be limited by the manufacturing technique. In case of hydroforming (see Section IV), the current limit is set by the machine table of the press and lies around 5 m for a blade segment. Also, for the field tests we wish to retrofit the
TABLE I. Main characteristics of a 1 kW demonstrator turbine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>2.4 m</td>
</tr>
<tr>
<td>Blade length</td>
<td>3.0 m</td>
</tr>
<tr>
<td>Blade chord</td>
<td>180 mm</td>
</tr>
<tr>
<td>Max. angular speed</td>
<td>240 rpm</td>
</tr>
</tbody>
</table>

blades onto an existing turbine, which has a rotor diameter of 2.4 m and a length of 3 m, with a 180 mm blade chord. The chord length and tip-speed ratio were chosen to optimise the power coefficient. This of course depends on the profile of the adopted aerofoil (see below). Together with the constraint on the centrifugal forces, the design tip-speed ratio sets the maximum angular velocity. A DMST-simulation based on the S2027-aerofoil (see below) and for a Reynolds number of $7 \times 10^5$ predicts a maximum power coefficient of 0.48 at a tip-speed ratio of 3.25. (Of course the true $C_P$ will be lower, as the presented values have not been corrected for tip effects, drag of the struts, friction losses and generator efficiency.) This leads to a wind turbine with an estimated rated power of 1 kW, with the dimensioning summarised in Table I.

![Image of power coefficient vs tip-speed ratio](image)

FIG. 2. Power coefficient $C_P$ as a function of tip-speed ratio, calculated with a DMST model for a Reynolds number of $7 \times 10^5$.

A large number of profiles have been suggested for adoption in VAWT rotors, see e.g. the classic SANDIA report\textsuperscript{18}. It has also been suggested\textsuperscript{19} that mildly asymmetric profiles
offer distinct advantages over symmetrical profiles. Based on our overview of the existing literature, we have selected the Selig S2027 profile for the present work (Fig. 3). The design goal of this profile\textsuperscript{20} is to have low drag in the range of lift coefficients required for high speed, while still maintaining reasonable performance at moderate Reynolds numbers. These results are in accordance with others like Guillaume, Algazze, and Duc\textsuperscript{21} where, in all computed configurations for a small VAWT, the S2027 aerofoil always gave better results than other common profiles.

![FIG. 3. The Selig S2027 profile.](image)

C. Adaptation of the selected aerofoil to metal sheet manufacturing

The very sharp trailing edges that are typical for aerodynamically optimal profiles are difficult (or too costly) to manufacture from sheet metal. This is also the case for composite materials, where in practice blades always have a small radius at the trailing edge.

When manufacturing a blade from sheet metal, different options are available for the trailing edge, depending on the position where the deformed blank starts and ends. The different options we considered are shown in Fig. 4. If the blank starts and ends at the trailing edge, there are two possible trailing-edge solutions: either both blank ends are cut off at the same position (case V1 in Fig. 4) or one blank end protrudes past the other end, creating a small notch. This notch may be filled during a subsequent welding process. The protruding end can either be on the blade’s down- or upside (cases V2 and V3, respectively), which makes a difference when using asymmetrical profiles. If the blank does not start and end at the trailing edge, it must be bent with a radius as small as possible at the trailing edge (V4). Usually, the minimum (inner) bending radius that can be achieved without damaging...
FIG. 4. Different trailing-edge shapes (left panel) and corresponding lift curves (right panel) computed from 2-D computational fluid dynamics. The top half of the right panel shows the lift curves, the lower half shows the maximum absolute difference between any two trailing-edge shapes, as a function of the angle of attack.

the blank is equal to the sheet thickness. As a result, the minimum outer trailing-edge radius is twice the thickness of the blank.

We have investigated the aerodynamic behaviour of the four different designs for the trailing edge by means of two-dimensional CFD simulations. The lift curve produced by each profile is shown in Fig. 4 (right panel, top half). The differences between the proposed geometries are small: as shown in Fig. 4 (right panel, lower half), the maximum differences in lift coefficient are smaller than 0.05, corresponding to relative differences smaller than 13%.

The best design for the trailing edge will be a compromise between manufacturing cost, aerodynamic performance, and noise. Trailing-edge noise is mainly caused by the interaction of the boundary layer with the trailing edge, by airflow separation, and by vortex shedding (for blunt trailing edges, referred to as bluntness noise). The sharp edges and irregular transitions of options V1, V2 and V3, will contribute significantly to such turbulent interactions and thus to noise. Also, the manufacturing process (hydroforming) might not work properly near sharp edges, since there is a high risk of leaks.

We have opted for a rounded trailing edge (V4), due to its simplicity for manufacturing
and the expected absence of turbulent noise. No further edge machining is required, and there are no appreciable aerodynamic differences. Moreover, design V4 allows for an inner reinforcement by folding in part of the sheet where the two edges meet and welding it against the profile on both the intrados and the extrados. As the trailing-edge thickness is comparable to the thickness of the boundary layer (between 6 mm at 5 m/s and 4.7 mm at 25 m/s using flat-plate boundary layer theory\textsuperscript{23}), a certain degree of tonal noise is to be expected. However, when upscaling the prototype to larger chord lengths, the profile becomes comparatively sharper (as the manufacturing constraints do not change) so that the production of tonal noise is expected to decrease.

The final geometry is shown in Fig. 5 (dashed line) and compared with the original shape (solid line).

![FIG. 5. Shape of the S2027 aerofoil adapted for sheet metal manufacturing (blue dashed line) compared with the original profile (black solid line).]

III. STRUCTURAL ANALYSIS OF VAWT BLADES

A. Blade loads

The aim of the structural analysis is to design the blade in such a way that the deformation and stresses caused by the loads on the blade are within an acceptable range. The main design parameters are the material choice, sheet thickness and internal reinforcements.

The main forces acting on a VAWT blade are shown in Fig. 6. The inertial force is the dominant cause of blade deformation and stress. Superimposed on this inertial base load are the aerodynamic forces, lift and drag, which cause a cyclically varying load at the rotational frequency of the rotor. Over the design lifetime of 20 years for a typical wind turbine, a rotor blade will undergo more than a hundred million load cycles, making fatigue resistance a primary design driver. Obviously, to allow Maximum Power Point Tracking (MPPT),
the rotor rpm will vary with wind speed, and thus also the inertial base load. This should be considered as well when studying fatigue.

![Diagram of aerodynamic forces](image)

**FIG. 6.** The aerodynamic force produced by the blade varies while it rotates due to the change in the angle of attack.

We categorise our prototype turbine as a class IV turbine according to the IEC 61400-2 design standard for small wind turbines\(^\text{24}\). This implies that our turbine is intended for moderate wind climates with average wind speeds of 6 m/s and a reference wind speed of 30 m/s. Here we focus on two load cases: extreme loads (at maximum rotational speed, and in extreme wind speeds in parked conditions) and fatigue load.

We performed a Finite-Element Analysis (FEA) using CATIA V5 of the blade section in between the struts (1.5 m long). To estimate the loads, we bounded the operating envelope of the turbine. The rotor speed varies with wind speed to achieve a tip-speed ratio of three, between cut-in (estimated at 5 m/s) till rated speed at 10 m/s. Beyond 10 m/s the rotational speed was limited to 240 rpm until cut-out, which we set at 25 m/s. The maximum equivalent stress that is expected in extreme operation (at 25 m/s wind and 240 rpm) is 151 MPa, with a corresponding deformation of 2.87 mm.

In parked conditions, the stress caused by the 50-year extreme wind speed (calculated according to the IEC standard\(^\text{24}\)) is only 37 MPa and thus small.

Also the fatigue behaviour was studied according to the IEC standard. We estimated the variation in aerodynamic and inertial load for all wind speeds between 5 m/s and 25 m/s. This is combined with the probability of occurrence of every wind speed over the 20 year lifetime of the turbine (assuming a Rayleigh distribution), to obtain the number of revolutions (i.e. load cycles of the aerodynamic forces) and the mean stresses (32.7 MPa to 133.7 MPa) and stress ranges (6.0 MPa to 34.8 MPa peak-to-peak). After correction for the
mean stress through the Goodman approximation, which is a conservative estimate\textsuperscript{25}, and
given the material properties as described in Section III C, we found that the Miner’s rule
accumulated damage\textsuperscript{24} was only 0.08, and thus well below 1.

Assessing the impact of the fluctuating inertial loads due to variations in rotor rpm is more
subtle. One needs to know the time scales of wind speed variations, the rotor inertia and the
time constant of the MPPT control architecture. However, we can safely neglect small wind-
speed variations (with small stress fluctuations) on the same ground as the above-studied
aerodynamic cycles. Furthermore, we estimate the number of start-stop cycles, with a stress
range of 0 to a maximum of 151 MPa, at $10^6$ (i.e. 5 times per hour, over 20 years). This
is well below the fatigue limit, where the stress-life curve flattens, of the chosen material,
as is shown in Fig. 7. (The blade should withstand more than $10^{10}$ cycles.) This should be
sufficient for our prototype blade design. For certification however, we expect that a more
thorough fatigue analysis with a full aeroelastic turbine model according to IEC 61400-2
will be required.

![Graph showing stress-life curve and maximum stress level encountered in simulations.]

FIG. 7. Stress-life curve of the chosen material (solid line), and maximum stress level encountered
in our simulations (dashed line).

B. Structural design of the blade

The stress values we calculated for the extreme and fatigue loads in Sec. III A, were ob-
tained for one specific (the final) blade configuration: a chord length of 180 mm, a blade
section length (between the struts) of 1.5 m, a sheet thickness of 1 mm with inner rein-
forcement. The stresses and deformations were obtained from FE simulations using solid
elements. It is important to realise that these stress and deformation levels depend critically on the blade configuration, as we will illustrate now. We simulate different configurations for rated conditions (10 m/s wind speed, 240 rpm), where the load is just shy of 90% of the maximum load. To test a multitude of different configuration, we used shell elements to speed up the FE computations. The precise values of deformation and stress will differ slightly from those of Sec. III A.

In the demonstrator turbine, the struts will be attached to the blade at 25% and 75% span. The structural simulation therefore only covers the part of the blade between the struts (50% of the total span); the outer sections of the blade behave quite similarly because of symmetry. In this design stage, a perfectly rigid clamping was assumed. As can be expected, the blade bends evenly over its length and shows its maximum deformation mid-span, near the trailing edge (see Fig. 8). The resulting maximum equivalent stresses appear at the fixed ends, usually where the profile is thickest, or at the contact points of a inner reinforcement, if present. The absolute maximum values, especially of the equivalent stress, may be overestimated due to the inelastic fixation of the blade ends in the simulation that will not appear in reality.

Using a safety factor of 1.5 (1.1 to account for uncertainty in the material, 1.35 in the load, according to the IEC 61400-2 standard\(^{24}\) and an ultimate stress of 530 MPa (see Section III C), we arrive at a maximum stress level of 350 MPa. In most cases we consider (e.g. Fig. 9) this leads to an unacceptably large deformation of the blade. For aerodynamic reasons we do not wish the blade to deform by more than a few mm. We therefore set the upper limit for deformation at 5 mm.

**FIG. 8.** Deformation (left) and maximum equivalent stress (right) at rated wind speed of 10 m/s and 240 rpm for 1.5 m span stainless steel 1.4404 blade with 1 mm thickness and an internal reinforcement.
Increasing the length of the blade section between the struts has a large effect on the deformation and equivalent stress, as shown in Fig. 9. It is not generally possible to decrease stresses and deformation by increasing the sheet thickness, as shown in Fig. 10 for a blade of 1.5 m span. From a certain thickness on, there is no further decrease in the maximum values of either stress or deformation. On the other hand, at very low sheet thickness, buckling effects result in an exponential increase of the maximum deformation and stress values. However, increasing the sheet thickness does allow an acceptable stress and deformation level to be maintained at larger chord lengths, as Fig. 11 shows.

![Graph showing influence of blade section length on deformation and equivalent stress.](image1)

**FIG. 9.** Influence of blade section length (for 1.5 mm sheet thickness and type 1.4404 steel) on the maximum deformation and equivalent stress.

To investigate the influence of reinforcements inside the blade, three different types have been compared to a blade without any reinforcements: tube, double T beam, and tailored blanks, see Fig. 12. (Tailored blanks are sheet metals with varying properties within the blank, e.g. different thicknesses, or specially hardened sections.) The FE simulations were done using solid elements (as we could not model the different reinforcements using shell elements) for a blade with 3 m section length and 1.5 mm thickness. Table II shows that an inner reinforcement not always reduces the maximum deformation of a blade, compare e.g. the tube with the blade without reinforcements. Only a double T beam results in a lower maximum deformation of the blade, but at the cost of increased maximum equivalent stress levels (at the connection of the reinforcement to the sheet). The same trend was observed for simulations of 1.5 m section length blades. These showed that double-T-beam-reinforced
Blades deform less (2.6 mm) but have higher stress levels (224 MPa) than blades without reinforcement (2.8 mm and 131 MPa respectively).

It turns out that finding the proper reinforcement requires a careful balance. On the one hand, the reinforcement should increase the resistance of the blade against bending. On the other hand, the reinforcement will increase the mass of the blade, and thus the centrifugal load. As bending resistance is proportional to the blade’s area moment of inertia, reinforcements help most when positioned as far as possible from the bending axis. Since
FIG. 12. Different internal reinforcements, from left to right: tube, double T beam, and tailored blanks.

TABLE II. Influence of reinforcement on maximum blade deformation and maximum equivalent stress, for a 3 m long blade with 1.5 mm sheet thickness.

<table>
<thead>
<tr>
<th></th>
<th>no reinforcement</th>
<th>tube</th>
<th>double T</th>
<th>tailored blanks</th>
</tr>
</thead>
<tbody>
<tr>
<td>deformation (mm)</td>
<td>49.5</td>
<td>51.5</td>
<td>47.6</td>
<td>49.3</td>
</tr>
<tr>
<td>stress (MPa)</td>
<td>575</td>
<td>690</td>
<td>836</td>
<td>585</td>
</tr>
</tbody>
</table>

the reinforcements are inside the blade, close to the bending axis, the increased resistance is counteracted by the increase in the centrifugal load, leading to a net effect that is often negative. For VAWT applications, a significant reduction of the maximum deformation values may only be achieved with stiffness increasing measures that do not increase the blade’s weight (e.g. honeycomb structuring).

For very thin blanks, an inner reinforcement by e.g. a perpendicular sheet may be required to reduce the differences in the deformation of the profile’s up- and downside, and thus maintain aerodynamic performance. We have therefore opted to use an inner reinforcement by making the sheet larger than the circumference of the profile and folding the excess length in the following manner: at the location on the intrados where the profile is closed, the excess length is folded inside the profile, first following the profile over a small fraction of the chord length, then crossing over to the extrados and again following the profile over a short length. This creates a reinforcement akin to an I-beam, but from the same sheet as the profile itself. This design allows for easy manufacturing and does not require any other manipulations than welding the contact lines (particularly no cutting). A FE simulation with solid elements for the final blade configuration shows a maximum deformation of 2.81 mm at 148 MPa for rated conditions.
TABLE III. Typical mechanical properties of the tested materials. $E$ is the modulus of elasticity, $\rho$ the density, $\sigma_u$ the ultimate tensile strength, and $\sigma_y$ the yield strength.

<table>
<thead>
<tr>
<th>Material</th>
<th>$E$ (GPa)</th>
<th>$\rho$ ($10^3$kg·m$^{-3}$)</th>
<th>$\sigma_u$ (MPa)</th>
<th>$\sigma_y$ (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.4404</td>
<td>200</td>
<td>7.98</td>
<td>530</td>
<td>220</td>
</tr>
<tr>
<td>Ti 1 RT 12</td>
<td>105</td>
<td>4.51</td>
<td>350</td>
<td>180</td>
</tr>
<tr>
<td>Al 5083</td>
<td>70</td>
<td>2.66</td>
<td>275</td>
<td>125</td>
</tr>
<tr>
<td>Mg AM20</td>
<td>45</td>
<td>1.75</td>
<td>200</td>
<td>90</td>
</tr>
</tbody>
</table>

C. Choice of the material for the blade

To determine the influence of different metal materials on the blade’s deformation behaviour, four metal types with varying sheet thickness and density have been compared: steel (stainless steel type 1.4404), titanium (Ti 1 RT 12), aluminium (Al 5083) and magnesium (Mg AM20). Some mechanical properties of these materials are given in Table III, while Fig. 13 shows their effect on the maximum deformation and stress of the blade.

![Graph showing influence of material choice and sheet thickness on the maximum blade deformation and equivalent stress.](image-url)

FIG. 13. Influence of material choice and sheet thickness on the maximum blade deformation and equivalent stress. The thick grey dashed line connects the points of equal blade weight.

From the structural analysis above, we conclude that the blade span between two struts should be limited to avoid excessive stresses. A length of 1.5 m gives reasonable values. Also a blade chord length of 180 mm is a good starting point that limits the equivalent stress in the blade. A sheet thickness of 1 mm seems to be a good compromise between blade mass,
deformation and equivalent stress. An inner reinforcement helps maintain an aerodynamic shape of the profile. Stresses and deformation can be brought down by using Mg AM20 rather than steel type 1.4404 (albeit with a lesser material strength). The possibility of using magnesium will be considered in the future, especially as Mg AM20 sheet metal is expected to become cheaper.

Once the design is adequate from the structural point of view, a new aerodynamic analysis should be carried out to ensure that the aerodynamic performance of the deformed shape under operational conditions is not degraded. We performed a CFD computation of the 3D blade for both the original and the maximum deformation geometry. Fig. 14 shows that the aerodynamic characteristics do not change much, even for the maximum deformation. Together with the above analysis this shows that the blade design is both structurally and aerodynamically sound.

![Polar curves for deformed and undeformed blade](image)

**FIG. 14.** Polar curves for deformed and undeformed blade (profile S2027, chord length 180 mm and blade length 1.5 m).

### IV. MANUFACTURING, RECYCLING, AND COST ANALYSIS OF VAWT BLADES

The value of current fibre-reinforced blades can add up to 30% of the whole turbine’s investment costs, where the turbine (tower included) typically makes up about 60% of the wind turbine project cost (foundation, grid connection and planning being other major contributors). Using metal materials and large-series production technology instead of manual
TABLE IV. Comparison of metal forming processes. We used CNC bending, roll forming, and hydroforming for the manufacturing of the blade.

<table>
<thead>
<tr>
<th></th>
<th>CNC bending</th>
<th>swing folding</th>
<th>roll rounding</th>
<th>roll forming</th>
<th>profile bending</th>
<th>extrusion forming</th>
<th>hydroforming</th>
</tr>
</thead>
<tbody>
<tr>
<td>eligibility for low quantities</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>eligibility for high quantities</td>
<td>−</td>
<td>0</td>
<td>−</td>
<td>++</td>
<td>−</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>variability (sizes)</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>−</td>
<td>−</td>
<td>−</td>
</tr>
<tr>
<td>investment cost</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>+</td>
<td>+</td>
<td>−</td>
</tr>
<tr>
<td>state of the art process</td>
<td>++</td>
<td>++</td>
<td>+</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>++</td>
</tr>
<tr>
<td>repeating accuracy</td>
<td>+</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>−</td>
<td>++</td>
<td>+</td>
</tr>
<tr>
<td>future potential (complex profiles)</td>
<td>−</td>
<td>0</td>
<td>−</td>
<td>++</td>
<td>/</td>
<td>+</td>
<td>++</td>
</tr>
</tbody>
</table>

processes of composites, offers a potentially large cost reduction. Moreover, metal blades are easier to recycle, reducing the total life-cycle impact of turbine blades.

Apart from the differences in material and manufacturing costs between metal and composite blades that will be described below, also recycling issues distinguish these materials. A full Life-Cycle Assessment (LCA) of metal blades and composite blades is beyond the scope of this paper. Yet some general conclusions can be drawn from the vast number of LCAs of entire turbines available in the literature, see e.g. Haapala and Prempreeda\(^{26}\) for a specific comparison of two turbines with composite blades and a steel tower, or Dolan and Heath\(^{27}\) for a summary of multiple LCA studies. Both steel and composites have the greatest (negative) impact at the level of raw material extraction and production. With respect to end-of-life treatment however, steel can be successfully recycled (losses of 10% are common) while composites are typically only used for landfill. Without more mature recycling techniques for composites, steel is advantageous from a cradle-to-grave perspective.

A. Choice of the manufacturing technology

Several metal forming methods were compared using criteria like eligibility for low and high quantities, size variability, repeating accuracy, invest and process cost as well as their potential for more complex rotor designs (Table IV).

When comparing such manufacturing processes, one has to consider four main factors that affect the production cost:
• General production parameters (e.g. quantity, lot sizes, production rates, makespan or production life cycle, depreciation time, interest rates),

• Investment costs such as machines, tooling, research and development (R&D), handling equipment,

• Material costs (depending on the specific alloy and the usage of semi-finished products, process specific rate of material use),

• Process costs (e.g. machine and man-hour rates for the manufacturing process).

Usually, the necessary equipment and machines for the production of blades is already available at contract manufacturers. Therefore the investment cost is often the part-specific tooling and the development and engineering of a production process. Depending on the number of parts that will be produced over the makespan of the part, their contribution to a part’s production cost varies widely.

The focus in this paper is on straight H-type VAWT blades with constant cross section, as a first step. Such blades are currently made as extruded aluminium parts or from fibre-reinforced composite materials. For such easy-to-manufacture blades, certainly in small to medium quantities, manufacturing processes should be based on flexible tooling technology without part-specific milled dies. Therefore we have chosen bending by Computer Numerical Control (CNC) as the optimal manufacturing process for the preform of the blade prototypes. In the long run, we work towards large-series production technologies that can be extended towards more complex rotor designs (curved blade axes, varying cross sections, surface structuring). Therefore, we have opted for a combination of roll forming and hydroforming for the series production technology.

Hydroforming always requires some type of preform, usually a straight or bent closed profile or tube, that is then formed with a high pressure (HP) medium (mainly liquid or gas, depending on the forming temperature) from the inside to the outside in a die of the final geometry. Processes that are technically suitable to produce such preforms for our case are CNC bending (resulting in an edgy preform profile) and roll forming (round profile).

The hydroforming process is controled by several parameters: the HP fluid’s inner pressure for forming the part (this depends on the material strength and thickness), the press closing force to keep the press closed (against the internal pressure), and the axial sealing
force to prevent leakage at the part’s ends. The hydroforming process itself contains four main steps:

- placing the preform in the opened press, and closing the press,
- pushing sealing stamps slightly into the profile ends, and filling the part with fluid while evacuating the remaining air,
- building up the press closing and the axial sealing forces while increasing the pressure of the fluid; this is the actual forming step,
- gradually releasing the internal pressure and the forces, opening the press and removing the formed part.

Due to the forming operations, stresses are induced in the material that can remain after the forming. The highest forming stresses for our design occur at the trailing edge, given the small curvature of the profile there. These stresses are not expected to cause durability issues, as the trailing edge is not very stressed. The load-bearing parts of the blade are much less curved and thus less prone to residual stresses. Yet, should future findings indicate that durability is affected, one could always envisage a post-forming heat treatment step to alleviate the stresses.

B. Cost benefits of metal blades

A large benefit of using metal materials is their low price per kilogram compared to fibre-reinforced materials. While Glass-Fibre-Reinforced Plastic (GFRP) can reach a cost of 7-8 €/kg (and Carbon-Fibre-Reinforced Plastics even 30-50 €/kg), steel alloys are well below that value\(^7\). Stainless steel sheet metals that may be considered for the prototype turbine usually cost between 4 €/kg and 5 €/kg and large series car-industry steels like DP600 are just under \(\sim 1\) €/kg but have to be treated against corrosion. Our 3 m long blade prototype (with sheet thickness of 1 mm) will weigh around 10 kg, resulting in a material cost of 40-50 €. This is about the same cost as for a GFRP blade, since such a blade would only weigh around 5-6 kg. CFRP would be substantially lighter but much more expensive (at around 200 €).
TABLE V. Breakdown of manufacturing costs, per blade, for metal and GFRP series production.

<table>
<thead>
<tr>
<th></th>
<th>Cost in €</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Metal prototypes</td>
</tr>
<tr>
<td>R&amp;D, Tooling cost</td>
<td>121</td>
</tr>
<tr>
<td>Material cost</td>
<td>43</td>
</tr>
<tr>
<td>Manufacturing cost</td>
<td>33</td>
</tr>
<tr>
<td>Total cost</td>
<td>197</td>
</tr>
</tbody>
</table>

The second benefit of metal over fibre-reinforced materials lies in the manufacturing process. Even if automated production methods for fibre reinforced parts are under development, some general process-specific disadvantages will remain. Compared to sheet-metal production with typical press cycle times of 5-30 seconds depending on the specific forming technique, the process of producing fibre-reinforced parts (laminating, vacuum injection, curing) consumes much more time (an order of magnitude or more) and is therefore more expensive.

The above-mentioned benefits are reflected in the order of magnitude difference in estimated production cost for metal versus fibre-reinforced blades. We assumed a production of 300 blades (of the design described before) over a makespan of one year for our prototype metal blades, and 3 000 blades per year over a makespan of five years for the large-series cost estimate (this for both metal and GFRP). We took into account the cost of the raw material, of the tooling required, and of the hydroforming press cost. The interest rate was set at 8%. The results are summarised in Table V. We find that cost for a metal blade is to 63 €, compared with 258 € for the GFRP blade produced in large series.

We conclude that the series-produced metal blades are more cost-efficient than GFRP blades starting from blade volumes of 1500-2000 per year. For low volumes, we suggest to employ the technology we used to construct prototype metal blades. This technology is more cost-efficient than GFRP series starting from 100-150 blades per year.

The additional blade mass makes careful balancing more important for our metal prototype than when lighter blades would be used. Also further research is needed to determine the most cost efficient structural reinforcements (foundation, tower, struts, connecting parts).
V. CONCLUSIONS

In this paper we show that it is possible to build metal blades for small and medium-sized wind turbines that are competitive with fibre-reinforced composite blades. The main parameters influencing the blade’s aerodynamic performance (operational parameters of the turbine, profile geometry) have been identified and evaluated taking into account structural effects (sheet thickness, inner reinforcements, materials, fatigue behaviour). The result is a first demonstrator blade and a basis for future potential through large-series production (e.g. for the use of magnesium, when its availability and level of examination increases and price decreases in the coming years).

The demonstrator blade (see Fig. 15) is 3 m in length (with 1.5 m bending length, which is the distance between the two struts) at a chord of 180 mm, made of stainless steel in 1.0 mm sheet thickness and with a perpendicular sheet as inner reinforcement. This blade has a moderate deformation and a tolerable stress level combined with a minimum amount of material and associated manufacturing costs.

![Prototype rotor prepared for the field tests.](image)

Through the use of mass production technology and performance optimization in a farm of counter-rotating turbines, it is possible to achieve as good a turbine performance as can be achieved with traditional composite blades while improving overall profitability and
VI. ACKNOWLEDGEMENTS

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