Extended calculations executed for the grid connected VIRYA-6.5 windmill

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KD 579

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1 Introduction

The VIRYA-6.5 windmill is meant for grid connection to a 50 Hz grid. The rotor geometry, the rotor characteristics and the matching of a 4-pole asynchronous generator for a certain gear ratio are calculated in report KD 578 (ref. 1). The P_{el}-V curves are given for two gear ratios i = 12.4 and i = 15.3 and provisionally i = 12.4 is chosen. The maximum electrical power is about 4.7 kW at a rated wind speed V_{rated} = 11 m/s. The design power is about 2.35 kW at a design wind speed V_d = 7 m/s. The cut in wind speed is about $V_{cut\ in}$ = 4.3 m/s.

The main advantage of this system is that the generator and the transmission are standard components which are rather cheap. The selected motor gear box combination of manufacture Rossi costs € 1.338 including transport and excluding VAT. An extra advantage of this system is that at high wind speeds, the rotor runs at a low tip speed ratio and that the aerodynamic noise production will therefore be limited.

The main disadvantages of this system are that the C_p of the rotor is only high at wind speeds around the design wind speed and that a special soft starter is required to connect the generator to the grid if the wind speed is around the cut-in wind speed. Some research has been done to find such soft starter which is designed for a rather low nominal power of 4 kW but up to now it could not be found.

It must be possible to stop the rotor. This can be done by a brake or by lifting the vane blade to the horizontal position. Provisionally it is chosen to use a standard brake on the generator shaft which can be supplied by Rossi. The extra costs of a spring loaded electromagnetic brake are € 328 which results in a total investment in the transmission and the generator of € 1.666.

Not all the calculations are given in report KD 578. This report KD 579 gives some extra calculations which are required to make the detailed drawings of the VIRYA-6.5 and to give the back ground for all the choices which are made.

2 Calculation of the strength of the spokes

The three blades are connected to each other by the spoke assembly. The spoke assembly is made of three spokes which are welded together under an angle of 120°. The spoke assembly is clamped in between the hub and a clamping disk and this prevents that the welds are loaded by a bending moment. A spoke has a length to the centre of 660 mm. The width b = 120 mm and the height h = 15 mm. The wooden blade has a width of 300 mm and a thickness of 44.5 mm and so the moment of resistance of the blade is much larger. It is therefore assumed that the spoke is the weakest component.

A spoke is loaded by a bending moment with axial direction which is caused by the rotor thrust and by the gyroscopic moment. A spoke is also loaded by a centrifugal force and by a bending moment with tangential direction caused by the torque and by the weight of the blade but the stresses which are caused by these loads can be neglected.

Because a spoke is rather thin it makes the blade connection elastic and therefore the blade will bend backwards already at a low load. As a result of this bending, a moment with direction forwards is created by a component of the centrifugal force in the blade. The bending is substantially decreased by this moment and this has a favourable influence on the bending stress.

It is started with the determination of the bending stress which is caused by the rotor thrust. There are two critical situations:

1^e The load which appears for a rotating rotor at V_{rated} = 11 m/s. For this situation the bending stress is decreased by the centrifugal moment. The yaw angle is 30° for V_{rated} = 11 m/s.

2^e The load which appears for a stopped rotor. The spoke strength is calculated if the rotor is stopped by a brake.

2.1 Bending stress in the spoke for a rotating rotor and V = 11 m/s

The rotor thrust is given by formula 7.4 of KD 35 (ref. 2). The rotor thrust is the axial load of all blades together and exerts in the hart of the rotor. The thrust per blade $F_{t\,\delta\,bl}$ is the rotor thrust $F_{t\,\delta}$ divided by the number of blades B. This gives:

$$F_{t\,\delta\,bl} = C_t * \cos^2\delta * ½\rho V^2 * \pi R^2 / B \qquad \text{(N)} \qquad (1)$$

For the rotor theory it is assumed that every small area dA which is swept by the rotor, supplies the same amount of energy and that the generated energy is maximised. For this situation the wind speed in the rotor plane has to be slowed down till 2/3 of the undisturbed wind speed V. This results in a pressure drop over the rotor plane which is the same for every value of r. It can be proven that this results in a triangular axial load which forms the thrust and in a constant radial load which supplies the torque.

The theoretical thrust coefficient C_t for the whole rotor is 8/9 = 0.889 for the optimal tip speed ratio. In practice C_t is lower because of the tip losses and because the blade is not effective up to the rotor centre. The effective blade length k' of the VIRYA-6.5 rotor is only 2.25 m but the rotor radius R = 3.25 m. Therefore there is a disk in the centre with an area of about 0.095 of the rotor area on which almost no thrust is working. This results in a theoretical thrust coefficient C_t = 8/9 * 0.905 = 0.804. Because of the tip losses the real C_t value is substantially lower. Assume this results in a real practical value of C_t = 0.7. It is assumed that the thrust coefficient is constant for values of λ in between 0.75 λ_d and $\lambda_{unloaded}$.
Substitution of C_t = 0.7, δ = 30°, ρ = 1.2 kg/m^3, V = 11 m/s, R = 3.25 m and B = 3 in formula 1 gives $F_{t\,\delta\,bl}$ = 422 N.

For a pure triangular load, the same moment is exerted in the hart of the rotor as for a point load which exerts in the centre of gravity of the triangle. The centre of gravity is lying at 2/3 R = 2.167 m. Because the effective blade length is only k', there is no triangular load working on the blade but a load with the shape of a trapezium as the triangular load over the part R – k' falls off. The centre of gravity of the trapezium has been determined graphically and is lying at about r_1 = 2.28 m.

The maximum bending stress is not caused at the hart of the rotor but at the edge of the hub because the strip bends backwards from this edge. This edge is lying at r_2 = 0.075 m. At this edge we find a bending moment $M_{b\,t}$ caused by the thrust which is given by:

$$M_{b\,t} = F_{t\,\delta\,bl} * (r_1 - r_2) \qquad \text{(Nm)} \qquad (2)$$

Substitution of $F_{t\,\delta\,bl}$ = 422 N, r_1 = 2.28 m and r_2 = 0.075 m in formula 2 gives $M_{b\,t}$ = 930 Nm = 930000 Nmm.

For the stress we use the unit N/mm^2 so the bending moment has to be given in Nmm. The bending stress σ_b is given by:

$$\sigma_b = M / W \qquad \text{(N/mm}^2\text{)} \qquad (3)$$

The moment of resistance W of a strip is given by:

$$W = 1/6\ bh^2 \qquad \text{(mm}^3\text{)} \qquad (4)$$

(3) + (4) gives:

$$\sigma_b = 6\ M / bh^2 \qquad \text{(N/mm}^2\text{)} \qquad \text{(M in Nmm)} \qquad (5)$$

Substitution of M = 930000 Nmm, b = 120 mm and h = 15 mm in formula 5 gives $\sigma_b = 207$ N/mm^2. For this stress the effect of the stress reduction by bending forwards of the blade caused by the centrifugal force in the blade has not yet been taken into account. The gyroscopic moment has also not yet been taken into account.

Next it is investigated how far the blade bends backwards as a result of the thrust load and what influence this bending has on the centrifugal moment. Hereby it is assumed that the strip is only bending over part from the inner connecting bolt to the hub. The inner connecting bolt point lies at r_3 = 0.295 m = 295 mm.

So the length of the strip l which is loaded by bending is given by:

$l = r_3 - r_2$ (mm) (6)

The load from the blade on the strip at r_3 can be replaced by a moment M and a point load F. F is equal to $F_{t\,\delta\,bl}$. M is given by:

$M = F * (r_1 - r_3)$ (Nmm) (7)

The bending angle ϕ (in radians) at r_3 for a strip with a length l is given by (combination of the standard formulas for a moment plus a point load):

$\phi = l * (M + ½ Fl) / EI$ (rad) (8)

The bending moment of inertia I of a strip is given by:

$I = 1/12\ bh^3$ (mm^4) (9)

(6) + (7) + (8) + (9) gives:

$\phi = 12 * F * (r_3 - r_2) * \{(r_1 - r_3) + ½ (r_3 - r_2)\} / (E * bh^3)$ (rad) (10)

Substitution of F = 422 N, r_3 = 295 mm, r_2 = 75 mm, r_1 = 2280 mm, E = 2.1 * 10^5 N/mm^2, b = 120 mm and h = 15 mm in formula 10 gives: ϕ = 0.02744 rad = 1.57°. This is an angle which can't be neglected. In report R409D (ref. 3) a formula is derived for the angle ε with which the blade moves backwards if it is connected to the hub by a hinge. This formula is valid if both the axial load and the centrifugal load are triangular. For the VIRYA-6.5 this is not exactly the case but the formula gives a good approximation. The formula is given by:

$$\varepsilon = \arcsin \left(\frac{C_t * \rho * R^2 * \pi}{B * A_{pr} * \rho_{pr} * \lambda^2} \right) \quad (°) \qquad (11)$$

In this formula A_{pr} is the cross sectional area of the airfoil (in m^2) and ρ_{pr} is the density of the used airfoil material (in kg/m^3). The average thickness of a Gö 711 airfoil is about a factor 0.7 of the maximum thickness. So the cross sectional area is 0.7 * 44.5 * 300 = 9345 mm^2 = 0.009345 m^2. The blade is made of hard wood with a density ρ_{pr} of about ρ_{pr} = 0.6 * 10^3 kg/m^3. In figure 4 of KD 578 it can be seen that for high wind speeds, the rotor is running at about λ = 4.5. Substitution of C_t = 0.7, ρ = 1.2 kg/m^3, R = 3.25 m, B = 3, A_{pr} = 0.009345 m^2, ρ_{pr} = 0.6 * 10^3 kg/m^3 and λ = 4.5 in formula 11 gives: ε = 4.69°. This angle is larger than the calculated angle of 1.57° with which the blade would bend backwards if the compensating effect of the centrifugal moment is not taken into account. This means that the real bending angle will be less than 1.57°.

The real bending angle ε is determined a follows. A thrust moment M_t = 930 Nm is working backwards and M_t is independent of ε for small values of ε. A bending moment M_b is working forwards and M_b is proportional with ε. M_b = 930 Nm for ε = 1.57°. A centrifugal moment M_c is working forwards and M_c is also proportional with ε. M_c = 930 Nm for ε = 4.69°. The path of these three moments is given in figure 1. The sum total of $M_b + M_c$ is determined and the line $M_b + M_c$ is also given in figure 1.

fig. 1 Path of M_t, M_b, M_c, and $M_b + M_c$ as a function of ε

The point of intersection of the line of M_t with the line of $M_b + M_c$ gives the final angle ε. In figure 1 it can be seen that ε = 1.18°. This is a factor 0.75 of the calculated angle of 1.57°. Because the bending stress is proportional to the bending angle it will also be a factor 0.75 of the calculated stress of 207 N/mm^2 resulting in a stress of about 156 N/mm^2. This is a rather low stress but up to now the gyroscopic moment, which can be rather large, has not yet been taken into account.

The gyroscopic moment is caused by simultaneously rotation of rotor and head. One can distinguish the gyroscopic moment in a blade and the gyroscopic moment which is exerted by the whole rotor on the rotor shaft and so on the head. On a rotating mass element dm at a radius r, a gyroscopic force dF is working which is maximum if the blade is vertical and zero if the blade is horizontal and which varies with sinα with respect to a rotating axis frame. α is the angle with the blade axis and the horizon. So it is valid that $dF = dF_{max} * \sin\alpha$. The direction of dF depends on the direction of rotation of both axis and dF is working forwards or backwards. The moment dF * r which is exerted by this force with respect to the blade is therefore varying sinusoidal too.

However, if the moment is determined with respect to a fixed axis frame it can be proven that it varies with $dF_{max} * r \sin^2\alpha$ with respect to the horizontal x-axis and with $dF_{max} * \sin\alpha * \cos\alpha$ with respect to the vertical y-axis. For two and more bladed rotors it can be proven that the resulting moment of all mass elements around the y-axis is zero.

For a single blade and for two bladed rotors, the resulting moment of all mass elements with respect to the x-axis is varying with $\sin^2\alpha$, so just the same as for a single mass element. However, for three and more bladed rotors, the resulting moment of all mass elements with respect to the x-axis is constant. The resulting moment with respect to the x-axis for a three (or more) bladed rotor is given by the formula:

$$M_{gyr\ x\text{-}as} = I_{rot} * \Omega_{rot} * \Omega_{head} \quad \text{(Nm)} \qquad (12)$$

In this formula I_{rot} is the mass moment of inertia of the whole rotor around the axis of rotation, Ω_{rot} is the angular velocity of the rotor and Ω_{head} is the angular velocity of the head.

The resulting moment is constant for a three bladed rotor because adding three $\sin^2\alpha$ functions which make an angle of 120° which each other, appear to result in a constant value. The three functions are given in figure 2. It can be proven for a three bladed rotor that the sum value of the three blades is equal to 3/2 of the peak value of one blade.

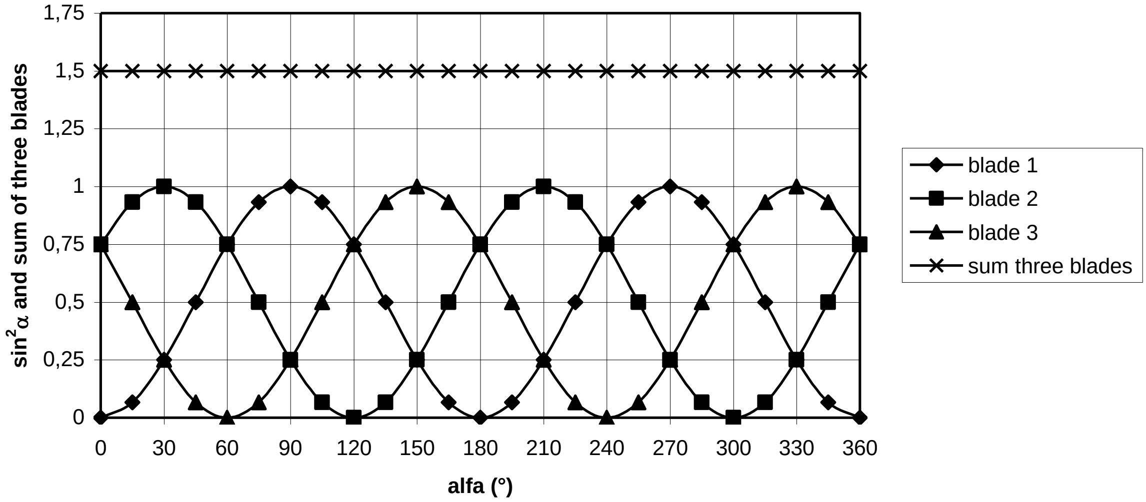

fig. 2 Path of $\sin^2\alpha$ and the sum of three blades

For the calculation of the blade strength we are not interested in the variation of the gyroscopic moment with respect to a fixed axis frame but in variation of the moment in the blade itself so with respect to a rotation axis frame for which it was explained earlier that the moment is varying sinusoidal. If the blade is vertical both axis frames coincide and the moment for both axis frames is the same. The maximum moment in one blade is then 2/3 of the sum moment as given by formula 12. The variation of the moment in the blade with respect to a rotating axis frame is therefore given by:

$$M_{gyr\ bl} = 2/3 \sin\alpha * I_{rot} * \Omega_{rot} * \Omega_{head} \quad \text{(Nm)} \qquad (13)$$

For a three bladed rotor, the moment of inertia of the whole rotor I_{rot} is three times the moment of inertia of one blade I_{bl}. Therefore it is valid that:

$$M_{gyr\ bl} = 2 \sin\alpha * I_{bl} * \Omega_{rot} * \Omega_{head} \quad \text{(Nm)} \qquad (14)$$

Up to now it is assumed that the blades have an infinitive stiffness. However, in reality the blades are flexible and will bend by the fluctuations of the gyroscopic moment. Therefore the blade will not follow the curve for which formula 13 and 14 are valid. I am not able to describe this effect physically but the practical result of it is that the strong fluctuation on the $\sin^2\alpha$ function is rather flattened. However, the average moment is assumed to stay the same as given by formula 14. I estimate that the flattened peak value is given by:

$$M_{gyr\ bl\ max} = 1.2 * I_{bl} * \Omega_{rot} * \Omega_{head} \quad \text{(Nm)} \qquad (15)$$

For the chosen blade geometry it is calculated that I_{bl} = 66 kgm². The maximum loaded rotational speed of the rotor can be read in figure 4 of KD 578 and it is found that n_{max} = 128 rpm. This gives $\Omega_{rot\ max}$ = 13.4 rad/s (because $\Omega = \pi * n / 30$).

It is not easy to determine the maximum yawing speed. The VIRYA-6.5 is provided with the hinged side vane safety system which has a light van blade and a large moment of inertia of the whole head around the tower axis. This is because the vane arm is a part of the head. For sudden variations in wind speed and wind direction the vane blade will therefore react very fast but the head will follow only slowly. It is assumed that the maximum angular velocity of the head can be 0.2 rad/s at very high wind speeds.
Substitution of I_{bl} = 66 kgm^2, $\Omega_{rot\ max}$ = 13.4 rad/s en $\Omega_{head\ max}$ = 0.2 rad/s in formula 15 gives: $M_{gyr\ bl\ max}$ = 212 Nm = 212000 Nmm.
Substitution of M = 212000 Nmm, b = 150 mm and h = 15 mm in formula 5 gives $\sigma_{b\ max}$ = 38 N/mm^2. This value has to be added to the bending stress of 156 N/mm^2 which was the result of the thrust because there is always a position were both moments are strengthening each other. This gives $\sigma_{b\ tot\ max}$ = 194 N/mm^2. The minimum stress is 156 - 38 = 118 N/mm^2. So the stress is not becoming negative and therefore it is not necessary to take the load as a fatigue load.

For the spoke material bare drawn (so with the rolling skin removed) mild steel (ST 37 2 K) is chosen. The 0.2 % deformation limit for this steel with a thickness of 15 mm is 300 N/mm^2. However, this is for a pulling stress. The deformation limit for a bending stress is higher and it is expected that it is about 400 N/mm^2. The calculated stress is much lower than the 0.2 % deformation stress for bending, so the strip is strong enough. In reality the blade is not extremely stiff and will also bend somewhat. This reduces the bending of the strip and therefore the stress is the strip will be somewhat lower than the calculated value.

2.2 Bending stress in the spoke for a stopped rotor

I is assumed that the rotor is stopped by a brake. For a stopped rotor there is no compensating effect of the centrifugal moment on the moment of the thrust. However, there is also no gyroscopic moment. The safety system is also working if the rotor is stopped but a much larger wind speed will be required to generate the same thrust as for a rotating rotor.

In chapter 2.1 it has been calculated that the maximum thrust on one blade for a rotating rotor is 422 N for V = V_{rated} = 11 m/s and δ = 30°. The head turns out of the wind such at higher wind speeds, that the thrust stays almost constant above V_{rated}. A stopped rotor will therefore also turn out of the wind by 30° if the force on one blade is 422 N. Also for a slowed down rotor the force is staying constant for higher yaw angles. However, for a stopped rotor, the resulting force of the blade load is exerting in the middle of the blade at r_4 = 1.75 m because the relative wind speed is constant along the whole blade. The bending moment around the edge of the hub is therefore somewhat smaller. Formula 2 changes into:

$$M_{b\ t} = F_{t\ \delta\ bl} * (r_4 - r_2) \quad \text{(Nm)} \qquad (16)$$

Substitution of $F_{t\ \delta\ bl}$ = 422 N, r_4 = 1.75 m en r_2 = 0.075 m in formula 16 gives $M_{b\ t}$ = 707 Nm = 707000Nmm. Substitution of M = 707000 Nmm, b = 120 mm and h = 15 mm in formula 5 gives σ_b = 157 N/mm^2. This is lower than the calculated stress for a rotating rotor. The load is not fluctuating and therefore it is surly not necessary to use the allowable fatigue stress. So the spoke is strong enough for a stopped rotor.

Because the spoke and the blade are rather flexible it has to be checked if a stopped rotor can't hit the tower. In chapter 2.1 it has been calculated, for no compensation of the gyroscopic moment, that the bending angle is 1.57° for a stress of 207 N/mm^2. So for a stress of 157 N/mm^2 the bending angle will be 1.57 * 157 / 207 = 1.19°. For a rotor radius of R = 3.25 m this results in a movement at the tip of about 0.068 m. Because the blade itself will bend too, the movement will be larger and it is expected that it will be about 0.15 m. The minimum distance in between the blade tip and the tower pipe is much larger if the blade is not bending. So there is no chance that the blade hits the tower for a stopped rotor.

3 Checking of the slow gear box shaft

The bending moment in the spokes due to the rotor thrust isn't resulting in a bending moment in the shaft if the thrust on all three blades is equal. However, the gyroscopic moment in the blades is transferred to the shaft. The gyroscopic moment in the shaft is given by formula 12. I_{rot} is three times I_{bl}, so $I_{rot} = 3 * 66 = 198$ kgm^2. Substitution of $I_{rot} = 198$ kgm^2, $\Omega_{rot} = 13.4$ rad/s and $\Omega_{head} = 0.2$ rad /s in formula 12 gives $M_{gyr} = 531$ Nm = 531000 Nmm.

The shaft has a diameter of 48 mm. The moment of resistance W of a shaft is given by:

$$W = \pi/32 * d^3 \text{ (mm}^3\text{)} \quad (17)$$

Substitution of d = 48 mm in formula 17 gives W = 10857 mm^3. Substitution of M = 531000 Nmm and W = 10857 in formula 3 gives that $\sigma_b = 49$ N/mm^2. This is a very low stress for a high quality steel shaft, so the shaft is strong enough.

A certain mass imbalance will also give a certain bending moment in the shaft but it is assumed that the rotor is well balanced and this bending moment therefore can be neglected.

The weight of the rotor will also give a certain bending moment. The shaft has a length of 110 mm. The mass of the rotor is about 100 kg so this gives a weight of about 1000 N. The centre of gravity lies a bit behind the front side of the shaft if the blades are not bending. But in reality, the blades are bending backwards a lot and it is assumed that the centre of gravity is lying at 55 mm from the collar on the shaft. So the bending moment M on this point is 1000 * 55 = 55000 Nmm. Substitution of this value and W = 10857 mm^3 in formula 3 gives $\sigma_b = 5$ N/mm^2 which is very low.

The slow gear box shaft is rather short. It has a double roller bearing at the front side and a single roller bearing at the back side. The allowable force F on the shaft depends on the product of the rotational speed and the number of hours L for which the bearings are loaded. Assume that the rotor is turning at the design rotational speed n = 123 rpm. Assume the wanted lifetime L is five years = 5 * 24 * 365 = 43800 hour. So L * n = 5387400.

The gyroscopic moment will also give a certain load on the bearings but this load is only active during fast movements of the head at high wind speeds. Therefore it is assumed that the gyroscopic moment can be neglected concerning the lifetime of the bearings and that the weight of the rotor is the only load which has to be taken into account.

In the Rossi catalogue a table is given for which the allowable load F at the middle of the shaft can be read as a function of the direction of the load and as a function of the torque. It is found that F = 3350 N for L * n = 5600000, for a torque of 280 Nm and for a downwards force for the given gear box position. The real design torque is about 227 Nm and the real force is about 1000 N, so the lifetime of the bearings will be much longer than 5 years.

4 Braking and grid connection of the generator

A standard spring loaded electromagnetic brake has a braking torque on the motor shaft of about 75 Nm. For a gear ratio i = 12.4, it means that the breaking torque at the slow gear box shaft is 75 * 12.4 = 930 Nm. The rotor torque is maximal for a design tip speed ratio $\lambda = 5$ (see KD 578 fig. 2). In figure 4 of KD 578 it can be seen that the mechanical power at $\lambda = 5$ and at a wind speed of 11 m/s is about 6700 W at a rotational speed of 140 rpm. Substitution of these values in formula 10 of KD 578 gives a maximum rotor torque of 457 Nm. So the braking torque of the brake on the motor shaft is certainly large enough to stop the rotor at any wind speed.

The brake is activated by a spring, so no power is used when the rotor is stopped. The brake is lifted by an electromagnet which is powered by a DC current which is gained by rectification of one of the 230 V phases. So lifting the brake will consume some power. The rotor will stop automatically when the grid falls off.

If the power consumption of the electromagnet isn't acceptable at low wind speeds when the rotor isn't producing any power, lifting of the brake should only take place above a 10 minutes average wind speed of about 4 m/s. So it is necessary to measure the wind speed and to activate the electromagnet as a function of the 10 minutes average wind speed. One has to take the 10 minutes average wind speed and not the momentary wind speed because the momentary wind speed varies a lot and this will result in unwanted braking.

The generator winding must be connected to the grid at a rotational speed of the generator which is only a very little higher than 1500 rpm. If the rotational speed is too high, the rotor will be slowed down too suddenly and this will cause a large peak torque in the transmission and also a large peak voltage on the grid. If the generator winding is connected to the grid, some current will flow even if the rotational speed is exactly 1500 rpm. This current requires a certain mechanical power of the rotor and this will slow down the rotor at a wind speed where the unloaded rotational speed of the generator is just 1500 rpm. But at a lower rotational speed than 1500 rpm, the generator will work as a motor and will take power from the grid. So if this happens, the connection in between the grid and the generator should be broken. To prevent fast connections and disconnections a so called soft starter is needed which gives some time delay in between the connection and disconnection. These soft starters have been developed for the old Danish grid connected windmills and I suppose that they are still available on the market. Some research has to be done to find a proper one.

The soft starter needs information about the rotational speed of the generator. This can be gained if the generator is provided by a separate tachometer. A separate tachometer can be supplied by Rossi but is rather expensive. May be it is possible to use the generator itself as a tachometer when it is running unloaded. The armature will have some remanent magnetism if it has been activated once. This magnetism produces a small voltage and the frequency of this voltage can be measured for an unloaded generator. If the voltage is too low, it might be possible to use four small circular permanent magnets which are glued in four small holes which are milled in the generator armature. The magnetic field of these magnets will be over powered by the rotating magnetic field of the stator so the frequency is always 50 Hz when the winding is connected to the grid. So disconnection can't be steered by the frequency. Disconnection must be activated if power is extracted from the grid. Another option might be to glue four magnets to the fan or to a rotating part of the brake and to measure the frequency by a separate coil. In this case the frequency can also be used for disconnection.

5 Checking of the head geometry

The head of the VIRYA-4.2 has been taken as starting point as this head has been tested for two years and it has functioned well. It is tried to scale this geometry up to the dimensions which are required for a rotor with a diameter of 6.5 m. The scale factor i = 6.5 / 4.2 = 1.548. If the relative strength of the head pipes must be the same it means that the moment of resistance must increase by a factor $1.548^3 = 3.71$. If the relative stiffness of the pipes must be the same it means that the moment of inertia must increase by a factor $1.548^4 = 5.74$.

The vane blade of the VIRYA-4.2 has a width and height of 1000 mm. If these sizes are scaled by a factor 1.548 the width and height would become 1548 mm. Plywood is available in The Netherlands in two standard sizes depending on the supplier. The most common sheet size is 4' * 8' = 1220 * 2440 mm. However, oucume plywood with sizes 1530 * 3100 mm is also available in several thicknesses. If such sheet is sawn in two identical parts an almost square vane blade with sizes 1530 * 1549 can be realised. The longest part is taken as height h. This vane blade has about the dimensions as required for a scale factor of 1.548.

The vane arm of the VIRYA-4.2 is build up from an inner part of 3" gas pipe with a length of 2 m and an outer part of 2" gas pipe with a length of 1.2 m, welded together using a tapered ring. Scaling of the length by a factor 1.548 would result in an inner length of 3.096 m and an outer length of 1.858 m, so in a total length of 4.954 m.

It is assumed that the vane arm of the VIRYA-6.5 can be build up from 3 m, 5” gas pipe and 1.5 m, 3” gas pipe so with a total length of 4.5 m. In this case both parts can be made from standard 6 m pipe without losses. The realised total length is shorter than according to scaling but this is compensated by taking a relatively smaller eccentricity than used for the VIRYA-4.2. The VIRYA-4.2 has an eccentricity e = 0.42 m and a diameter D = 4.2 m and so a ratio e / D = 0.1.

Next it is checked if 5” and 3” pipes are strong and stiff enough. The outside pipe diameter D_p, the wall thickness t, the inside pipe diameter d_p and the calculated values for W and I for the three given pipe values are given in table 1. d_p is given by:

$$d_p = D_p - 2 * t \qquad (mm) \qquad (18)$$

W and I can be calculated by:

$$W = \pi/32 * (D_p^4 - d_p^4) / D_p \qquad (mm^3) \qquad (19)$$

$$I = \pi/64 * (D_p^4 - d_p^4) \qquad (mm^4) \qquad (20)$$

	D_p (mm)	t (mm)	d_p (mm)	W (mm^3)	I (mm^4)
2” gas pipe	60.3	3.65	53	8679	261669
3” gas pipe	88.9	4.05	80.8	21907	973775
5” gas pipe	139.7	5.0	129.7	68796	4805412

Table 1 Values of D_p, t, d_p, W and I for 2”, 3” and 5” gas pipes

From the values of W and I, given in table 1, the ratios of W and I in between different pipes can be calculated. It is found that:

$W_{3”} / W_{2”}$ = 21907 / 8679 = 2.52 This is a factor 0.68 of 3.71
$I_{3”} / I_{2”}$ = 973775 / 261669 = 3.72 This is a factor 0.65 of 5.74
$W_{5”} / W_{3”}$ = 68796 / 21907 = 3.14 This is a factor 0.85 of 3.71
$I_{5”} / I_{3”}$ = 4805412 / 973775 = 4.93 This is a factor 0.86 of 5.74

If the calculated ratios of the real pipes are compared with the required values which are needed for the scale factor i = 1.548, it can be seen that the real factors are smaller, especially in between the 3” and the 2” pipes. But this effect is not as bad as it seems because the outer part is taken much shorter than according to the scale laws (1.5 m in stead of 1.854 m). The inner pipe is taken only a little shorter than according to the scale laws (3 m in stead of 3.096 m).

I think that the strength of the vane arm is not critical but that only the stiffness might be critical because the vane arm + vane blade may flutter at high wind speeds if the vane arm is not stiff enough. The stiffness of the whole vane arm is mainly determined by the inner 5” pipe and the relative stiffness of the 5” pipe is only a factor 0.86 lower than for the VIRYA-4.2. So I think that the chosen pipe diameters are correct in terms of the required strength and stiffness.

The next thing which has to be chosen is the eccentricity e. The eccentricity has to be chosen such that the rotor is about perpendicular to the wind direction for low wind speeds. The eccentricity should not be taken too small otherwise the self orientating moment will have a too large influence on the total rotor moment. For very low wind speeds the vane blade is in the almost vertical position and the balance of moments around the tower axis is then given by formula 49 or 50 of KD 223. Formula 50 is copied as formula 21.

$C_n = \pi R^2 * C_t * e / \{h * w * (R_v + i_1)\}$ (-) (21)

C_n is the normal coefficient of a square plate. The C_n-α curve of a square plate is given in figure 6 of KD 213 (ref. 4) or in figure 2 KD 551 (ref. 5). If the rotor is perpendicular to the wind, the angle α in between the wind direction and the vane blade is 30°. In figure 6 of KD 223, it can be read that C_n = 1.38 for α = 30°.

R is the rotor radius which is 3.25 m for the VIRYA-6.5. C_t is the thrust coefficient which is about 0.7 for a rotor with wooden blades with a Gö 711 airfoil. e is the eccentricity (m). It is chosen that e = 0.58 m. For this value of e it is found for the ratio e / D in between the eccentricity e and the rotor diameter D, that e /D = 0.089. This ratio is large enough to realise a stable functioning of the safety system.

h is the height of the vane blade and w is the width. h = 1.549 m and w = 1.53 m. R_v is the distance in between the hart of the tower and the leading edge of the vane blade measured in parallel to the vane axis. A composite drawing of the head is made to determine R_v. R_v depends on the position of the pin around which the head is turning in the head bearing housing. The position of the pin depends on the height H of the gear box and of the thickness of the gearbox bracket. The height of a gear box of Rossi size 100 is 200 mm. The thickness of the gear box bracket is chosen 10 mm. It is found that R_v = 3.88 m for this distance (see figure 4).

i_1 is the distance in between the normal force N acting on the vane blade and the leading edge. i_1 depends on the angle of attack α but is about 0.37 * w for α = 30° (see figure 7 KD 223 or figure 3 of report KD 551. So i_1 = 0.566 m for w = 1.53 m and $R_v + i_1$ = 4.446 m. Substitution of all these values in formula 21 gives that C_n = 1.28 .

In figure 6 of KD 223 (or figure 2 of KD 551) it can be seen that C_n = 1.28 belongs about to α = 28°. This angle is 2° smaller than the angle α = 30° for which the rotor is perpendicular to the wind. This means that the rotor makes a negative yaw angle δ = -2° with the wind direction for very low wind speeds. This is correct because in this case the rotor will be about perpendicular to the wind for a wind speed of about 6 m/s.

The end of the 3” pipe is flattened up to 15 mm wide gap. In this gap a steel strip with dimensions 100 * 15 * 1500 mm is welded. This strip makes a backwards angle of 15° with the 3” pipe. The vane blade is connected to this strip by four 3” stainless steel door hinges. In between the hinges, three sheets size 300 * 300 * 5 mm are bolted to the upper side of the strip. These sheets are bent such that they make a downwards angle of 3° with the horizon. These sheets function as an elastic stop for the vane blade and prevent that the angle in between the vane blade and the wind direction can become negative at very high wind gusts. This prevents flutter of the vane arm because the aerodynamic force on the vane blade can’t become negative for fast movements of the vane blade.

A steel pin with a diameter of 60 mm is welded in the 5” pipe at a distance of 523 mm from the hart of the 45° bevelled side. This pin contains a central hole of 20 mm for the generator wires. All VIRYA windmills use a stainless steel pin which turns in INA Permaglide bearings but one can also design a bearing housing with ball bearings. In this case it is not necessary to use stainless steel for the pin. The construction of the head bearing housing is not yet specified.

The gear box bracket is welded such to the 5” pipe that the rotor shaft makes a tilting angle of 5° with the vertical. This angle makes that the distance in between the blade tip and the tower is large enough to prevent that the blades may touch the tower at high wind gusts. Figure 4 is drawn without a tilting angle.

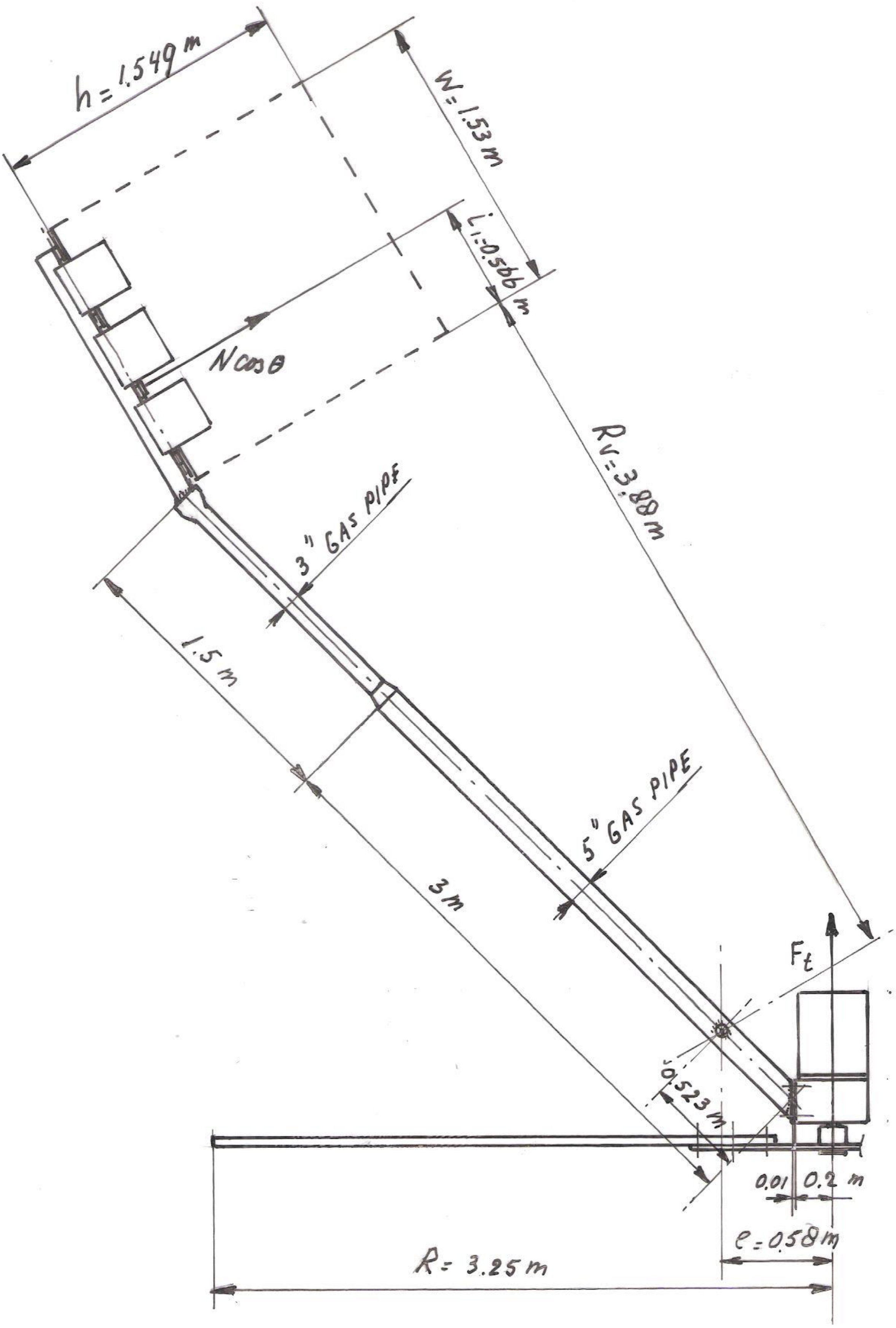

fig. 4 Top view head VIRYA-6.5 windmill

6 Checking of the strength of the tower

6.1 General

In KD 578 (ref. 1) it is written that the tower will be derived from the 12 m three legs tower of the VIRYA-4.2 or from the 8.4 m tubular tower of the VIRYA-3.3S. A three legs tower of a certain height will be lighter than a tubular tower of the same height but manufacture and painting or galvanising of a tubular tower is much easier. Provisionally a free standing tubular tower is chosen.

The rotor diameter of the VIRYA-6.5 is about a factor two larger than the rotor diameter of the VIRYA-3.3S and therefore it is decided to scale the VIRYA-3.3S tower up with about a factor 2. This means that it will get a height of about 16.8 m which seems acceptable. It will be made out of three 6 m sections which are bolted together. The overlap in between two sections is 0.6 m.

For the VIRYA-3.3S tower it was chosen to use 3” pipe with an outside diameter D_p = 88.9 mm and a wall thickness t = 2.5 mm for the upper section, 4” pipe with an outside diameter D_p = 114.3 mm and a wall thickness t = 2.75 mm for the middle section and 5” pipe with an outside diameter D_p = 139.7 mm and a wall thickness t = 3 mm for the lower section. The wall thickness was chosen as small as possible to minimize the tower weight.

If these values are scaled with a factor 2, the upper section will have dimensions of D_p = 177.8 mm and t = 5 mm, the middle section will have dimensions of D_p = 228.6 mm and t = 5.5 mm and the lower section will have dimensions of D_p = 279.4 and t = 6 mm. Not all these calculated dimensions are available in practice. Provisionally, the following pipes have been selected from the catalogue of the Dutch supplier Van Leeuwen Buizen.

Upper pipe: D_p = 177.8 and t = 5 mm, so the nominal inside diameter d_p = 167.8 mm.
Middle pipe: D_p = 244.5 and t = 6 mm, so the nominal inside diameter d_p = 232.5 mm.
Lower pipe: D_p = 298.5 and t = 7.1, so the nominal inside diameter d_p = 283.3 mm.

The pipes will be connected to each other in the same way as it is done for the VIRYA-3.3S tower. The construction is described for the joint in between the upper and the middle section but is similar for the connection of the middle and the lower section.

Two 30 mm thick rings are made with an inside diameter equal to the outside diameter of the upper pipe and an outside diameter equal to the inside diameter of the middle pipe. The lower ring is welded at the bottom side to the upper pipe. The upper ring is glued to the upper pipe by epoxy glue. Each ring is bolted to the inside of the middle pipe by four bolts M16 * 30 under 90°. The bending moment in the upper pipe is zero at the lowest ring so it is no problem if the weld causes stress concentration. The bending moment in the upper pipe is maximal at the upper ring but glue is not causing stress concentration. It even spreads the radial load on the ring very smoothly and it prevents a gap in between ring and pipe. The weight of the head is taken by the weld of the lower ring and by the four lower bolts.

The connection of the middle pipe to the lower pipe is similar. For the connection of the lower pipe to the foundation strips, a similar construction is chosen as for the connection of the pipe segments.

Two square sheets are made with dimensions 30 * 350 * 350 mm. Each sheet is provided with a central hole equal to the outside diameter of the lower pipe. Two threaded holes M20 mm holes are made at both opposite sides of the sheet at a pitch of 290 mm. The lower sheet is welded at the bottom side to the lower pipe. The upper sheet is glued to the lower pipe by epoxy glue. The foundation strips of the VIRYA-3.3S have a length of 1.2 m. Scaling with a factor 2 would give a length of 2.4 m but 3 m is chosen to prevent losses. The block of concrete is relatively longer but slenderer. The strip size is 15 * 350. Two 350 mm long distance rods which are cast in the concrete of the foundation keep the strips at a distance of 350 mm from each other during pouring of the concrete.

The two square sheets are clamped in between the two foundation strips by eight bolts M20 * 80. Two bolts can be used as a hinge for erection of the tower. One needs an auxiliary tower to do this and a winch. One has to use an auxiliary rope to prevent that the tower falls down in the wrong direction once it has reached the vertical position. The auxiliary rope can be removed once all eight bolts are tightened.

6.2 Calculations of the tower strength

For checking of the tower strength it is necessary to know the tower load. The tower top is loaded by a force F_{top} which is caused by the rotor thrust and by the aerodynamic force working on the vane arm and the vane blade. A moment works on the tower top which is caused by not being in balance of the vane weight and the rotor + generator weight but this moment is neglected. The tower is also loaded by the drag force working on each tower section. The forces for the upper, the middle and the lower section are called F_u, F_m and F_l.

F_{top} is mainly caused by the rotor thrust F_t. F_t is limited by the safety system because the rotor turns out of the wind at high wind speeds. It is assumed that the rotor turns out of the wind such that F_t is constant for V > 11 m/s. The yaw angle δ at V = 11 m/s is 30° (see KD 578 figure 3). The thrust at a yaw angle δ, $F_{t\delta}$ is given by formula 7.4 of report KD 35 (ref. 2). This formula is copied as formula 22.

$$F_{t\delta} = C_t * \cos^2\delta * ½\rho V^2 * \pi R^2 \quad \text{(N)} \qquad (22)$$

C_t is the thrust coefficient (-). The theoretical value is 8/9 = 0.89 but the real value is lower because of tip losses and because the effective blade length is shorter than R. Assume C_t = 0.7. Assume δ = 30° for V = 11 m/s. ρ is the air density which is about 1.2 kg/m^3 for air of 20° C at sea level. R is the rotor radius and R = 3.25 m for the VIRYA-6.5. Substitution of these values in formula 22 gives that $F_{t\delta}$ = 1265 N.

F_{top} is larger than $F_{t\delta}$ because there are also aerodynamic forces working on the vane arm and the vane blade. During wind gusts $F_{t\delta}$ may also be larger than the calculated value. Assume F_{top} = 1800 N.

The drag forces F_u, F_m and F_l are not reduced by the safety system. It is assumed that each force attaches to the middle of its tower section. These forces are maximal for the highest wind speed which may ever be expected. It is assumed that V_{max} = 35 m/s at the tower top. Because of the wind sheer, it is unrealistic to calculate the whole tower for this wind speed. It is assumed that V = 34 m/s for F_u, that V = 31 m/s for F_m and that V = 26 m/s for F_l. The drag force F is given by:

$$F = C_d * ½\rho V^2 * D_p * l \quad \text{(N)} \qquad (23)$$

C_d is the drag coefficient (-) which is 1.18 for smooth pipes if the Reynolds value is lower than 10^5 (see KD 213 figure 10, ref. 4). The Reynolds values have been calculated for each of the three sections and for the chosen maximum wind speeds and it was found that Reynolds is about 4.5 * 10^5. The drag coefficient for this Reynolds value is reduced to about 0.6 (see KD 213 figure 10). D_p is the outside pipe diameter in m. l is the visible pipe length in m.

Substitution of C_d = 0.6, ρ = 1.2 kg/m^3, V = 34 m/s, D_p = 0.1778 m and l = 5.4 m in formula 23 gives that F_u = 400 N.
Substitution of C_d = 0.6, ρ = 1.2 kg/m^3, V = 31 m/s, D_p = 0.2445 m and l = 5.4 m in formula 23 gives that F_m = 457 N.
Substitution of C_d = 0.6, ρ = 1.2 kg/m^3, V = 26 m/s, D_p = 0.2985 m and l = 5.4 m in formula 23 gives that F_l = 392 N.

A picture of the tower is given in figure 5. The forces F_{top}, F_u, F_m and F_l are given in this figure. The relevant dimensions are also given in figure 5. The tower has three critical cross sections U, M and L which are lying at the upper rings and at the upper square sheet. The bending moment M is calculated for each critical cross section and the bending stress σ is calculated using the formula 3. W out of formula 3 is the moment of resistance of the concerning pipe. W can be calculated if the outside and the inside pipe diameter is known but W may also be given in the catalogue of the pipe supplier. σ is calculated in N/mm^2 so for M the moment is taken in Nmm and for W the moment of resistance is taken in mm^3.

The main characteristics of the chosen pipes are given in table 2. The outside pipe diameter is called D_p (mm). The wall thickness is called t (mm). The inside pipe diameter is called d_p (mm). The moment of resistance is called W (mm^3). The moment of inertia is called I (mm^4). W and I are calculated using formulas 19 and 20. m is the pipe mass per meter. m_s is the pipe mass of a 6 m long section (excluding the rings).

pipe	D_p (mm)	t (mm)	d_p (mm)	W (mm^3)	I (mm^4)	m (kg/m)	m_s (kg)
upper	177.8	5	167.8	114057	9699721	21.2	127.2
middle	244.5	6	232.5	261639	31985346	35.1	210.6
lower	298.5	7.1	284.3	462520	69031073	50.7	304.2

Table 2 Characteristics of the chosen upper, middle and lower pipes

The total mass of the three pipes is 127.2 + 210.6 + 304.2 = 642 kg which seems acceptable for a tubular tower with a height of 16.8 m. The bending stress is now calculated for the cross sections U, M and L. The bending moment M_U is given by:

$$M_U = F_{top} * l_1 + F_u * l_2 \quad \text{(Nmm)} \tag{24}$$

Substitution of F_{top} = 1800 N, l_1 = 5400 mm, F_u = 400 N and l_2 = 2700 mm in formula 24 gives that M_U = 10800000 Nmm. Substitution of M_U = 10800000 Nmm and W = 114057 mm^3 in formula 3 gives that σ_U = 95 N/mm^2.

The bending moment M_M is given by:

$$M_M = F_{top} * (l_1 + l_3) + F_u * (l_2 + l_3) + F_m * l_4 \quad \text{(Nmm)} \tag{25}$$

Substitution of F_{top} = 1800 N, l_1 = 5400 mm, l_3 = 5400 mm, F_u = 400 N, l_2 = 2700 mm, F_m = 457 N and l_4 = 2700 mm in formula 25 gives that M_M = 23913900 Nmm. Substitution of M_M = 23913900 Nmm and W = 261639 mm^3 in formula 3 gives that σ_M = 92 N/mm^2.

The bending moment M_L is given by:

$$M_L = F_{top} * (l_1 + l_3 + l_5) + F_u * (l_2 + l_3 + l_5) + F_m * (l_4 + l_5) + F_l * l_6 \quad \text{(Nmm)} \tag{26}$$

Substitution of F_{top} = 1800 N, l_1 = 5400 mm, l_3 = 5400 mm, l_5 = 5400 mm, F_u = 400 N, l_2 = 2700 mm, F_m = 457 N, l_4 = 2700 mm, F_l = 392 N and l_6 = 2700 mm in formula 26 gives that M_L = 39320100 Nmm. Substitution of M_L = 39320100 Nmm and W = 464520 mm^3 in formula 3 gives that σ_M = 85 N/mm^2.

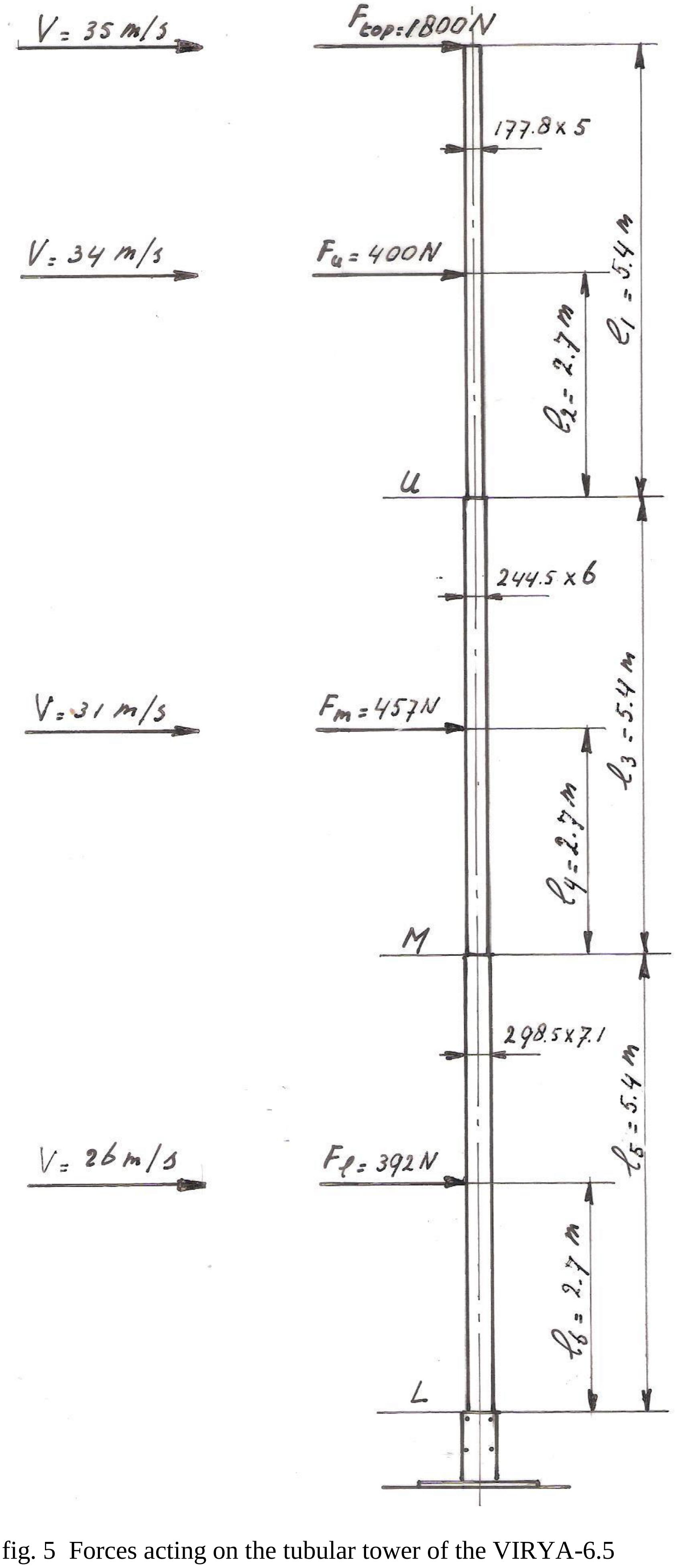

fig. 5 Forces acting on the tubular tower of the VIRYA-6.5

So the bending stress in cross sections U and M is about the same and about 93 N/mm^2. The bending stress in cross section L is somewhat lower and 85 N/mm^2. This calculation shows that the chosen pipe diameters and wall thicknesses are almost optimal for the VIRYA-6.5. The pipes are made of mild steel St37 (Fe 360). The allowable pulling stress for this material is about 180 N/mm^2 if the load is not a fatigue load. However, the allowable bending stress is higher than the allowable pulling stress and is about 240 N/mm^2. So the pipes are strong enough and have even a large reserve.

The performance of the tower is not only determined by the required strength but also by the required stiffness. The stiffness in combination with the pipe masses and the total mass of the head, the generator and the rotor determines the natural frequency. The first harmonic of the natural frequency must be about 1.4 Hz. In this case an unloaded rotor will go through this natural frequency at a wind speed of about 3 m/s. The tower will shake a little at this wind speed because the rotor will always have some mass imbalance or some aerodynamic imbalance but the energy in the oscillation is not high enough at 3 m/s to cause any damage. I can calculate the natural frequency for a tower made out of one pipe but for a tower made out of three sections the calculation is too complicated for me. So a prototype has to be built and the natural frequency has to be measured. This can easily be done by connecting a rope to the tower top and by bring the tower in oscillation. The number of oscillations per minute can be counted and then divided by 60 to find the frequency in Hz.

7 Use of the VIRYA-6.5 with a PM-generator

The main disadvantages of using an asynchronous generator is that the C_p of the rotor is only high at wind speeds around the design wind speed and that a special soft starter is needed to connect the generator to the grid. The main advantages of this system are that the whole transmission is simple and cheap and that an asynchronous generator has almost no sticking torque when it is not yet connected to the grid. This results in a low starting wind speed.

It might be possible to replace the asynchronous generator by a PM-generator with a 3-phase rectifier and use it for battery charging or for grid connection with a 3-phase inverter. In this case it is possible to follow the optimum cubic line of the rotor and the power output will be high at all wind speeds. However, a PM-generator with iron in the coils has a certain sticking torque and this may result in a too high starting wind speed. A 4-pole PM-generator with frame size 112 has been designed for the VIRYA-4.2 windmill and this generator has been measured for different loads for rotational speeds up to 1500 rpm. The measurements are given in report KD 200 (ref. 6).

The VIRYA-4.2 generator makes use of a special 5.5 kW motor. The stator stamping of this motor has a length of 150 mm compared to a length of 120 mm for a normal 4 kW motor. The armature has a length of 161 mm and has four inclined grooves. So the armature juts out 5.5 mm at each side of the stator. Four neodymium magnets size 40 * 30 * 10 mm are glued in each groove.

First it will be checked if this generator is strong enough to follow the optimum cubic line of the rotor up to the point of maximum power at a wind speed of 11 m/s. The optimum cubic line in the P-n graph corresponds to a parabolic line in the Q-n graph and so the point of maximum power corresponds to the point of maximum torque. In figure 4 and table 2 of KD 578 it can be seen that the maximum power is 7401 W at a rotational speed of 167.9 rpm. Next it is assumed that the gear box efficiency is 0.95 for all working points. So the mechanical power at the generator shaft is 0.95 * 7401 = 7031 W. The rotational speed for a gear ratio i = 12.4 is 12.4 * 167.9 = 2082 rpm. The torque at this rpm can be calculated using formula 10 of KD 578 (ref. 1). It is found that $Q = 30 * 7031 / (\pi * 2082) = 32.2$ Nm.

The generator has been measured for different resistances of the load for rectification in star up to a rotational speed of 1500 rpm. The maximum torque level at 2082 rpm can be determined by extrapolation. It is found that the maximum torque level is about 57 Nm (see figure 16 KD 200).

So the generator is strong enough to follow the optimum cubic line up to the point of maximum power and maximum torque, if the resistance of the load has the correct value. The generator efficiency depends on the load but is rather high and in between 75 % and 85 % for most loads (see figure 19, KD 200).

It might be possible to use a PM-generator which is made from a standard 4 kW motor with a stator length of 120 mm and an armature with a length of 121 mm and three magnets size 40 * 30 * 10 mm per groove. If it is assumed that the torque level decreases proportional with the stator length, it will decrease by a factor 120 / 150 = 0.8 and so the maximum torque level will be 45.6 Nm which is still high enough. The VIRYA-4.2 generator has a 35 mm tapered shaft and a foot B3. The generator drawing has to be changed anyway to give it a normal cylindrical 28 mm shaft with key groove and a flange for connection to the gear box.

The next point which has to be checked is the increase of the starting wind speed due to the sticking torque of the generator. For the time being it is chosen to use the measured characteristics of the original VIRYA-4.2 generator with a stator length of 150 mm. The measurements of the unloaded torque are given in figure 1 of KD 200 for star and for delta rectification. The rise of the unloaded torque at increasing rotational speed is much faster for delta rectification than for star rectification. The reason is that higher harmonic currents can circulate in the winding for delta rectification. Therefore star rectification is chosen. However, even for star rectification the unloaded torque is rising rather fast. For direct drive use in the VIRYA-4.2, this rise isn't faster than the rise of the rotor torque for the starting wind speed. However, if the generator is driven by an accelerating gear box, a small rise in rotor speed results in a very large increase in generator speed. This effect on the starting wind speed is investigated in the Q-n graph of the rotor. The Q-n curves of the rotor for different wind speeds are determined in the same way as the P-n curves but one has to use the formula for the torque Q in stead of the formula for the power P and one has to use the C_q-λ curve in stead of the C_p-λ curve. The C_q-λ curve is given in figure 2 of KD 578. For the rotational speed n, the same formula is used. The formulas for n and Q are copied from report KD 35.

Substitution of R = 3.25 m in formula 7.1 of KD 35 gives:

$$n = 2.9382 * \lambda * \cos\delta * V \qquad \text{(rpm)} \qquad (27)$$

Substitution of $\rho = 1.2$ kg / m^3 en R = 3.25 m in formula 7.7 of KD 35 gives:

$$Q = 64.707 * C_q * \cos^2\delta * V^2 \qquad \text{(W)} \qquad (28)$$

For a certain wind speed, for instance V = 3 m/s, related values of C_q and λ are substituted in formula 27 and 28 and this gives the Q-n curve for that wind speed. The calculated values of n and Q are given in table 3.

		V = 3 m/s δ = 0°		V = 4 m/s δ = 0°		V = 5 m/s δ = 0°		V = 6 m/s δ = 0°		V = 7 m/s δ = 0°		V = 8 m/s δ = 4.5°		V = 9 m/s δ = 13°		V = 10 m/s δ = 21.5°		V = 11 m/s δ = 30°	
λ	C_q	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)	n (rpm)	Q (Nm)
0	0.0089	0	5.2	0	9.2	0	14.4	0	20.7	0	28.2	0	36.6	0	44.3	0	49.9	0	52.3
1	0.011	8.8	6.4	11.8	11.4	14.7	17.8	17.6	25.6	20.6	34.9	23.4	45.3	25.8	54.7	27.3	61.6	28.0	64.6
2	0.02	17.6	11.6	23.5	20.7	29.4	32.4	35.3	46.6	41.1	63.4	46.9	82.3	51.5	99.5	54.7	112.0	56.0	117.4
3	0.0433	26.4	25.2	35.3	44.8	44.1	70.0	52.9	100.9	61.7	137.3	70.3	178.2	77.3	215.5	82.0	242.5	84.0	254.3
4	0.0675	35.3	39.3	47.0	69.9	58.8	109.2	70.5	157.2	82.3	214.0	93.7	277.8	103.1	335.9	109.4	378.1	112.0	396.4
5	0.078	44.1	45.4	58.8	80.8	73.5	126.2	88.1	181.7	102.8	247.3	117.2	321.0	128.8	388.1	136.7	436.9	140.0	458.0
6	0.0717	52.9	41.8	70.5	74.2	88.1	116.0	105.8	167.0	123.4	227.3	140.6	295.1	154.6	356.8	164.0	401.6	167.9	421.0
7	0.0557	61.7	32.4	82.3	57.7	102.8	90.1	123.4	129.8	144.0	176.6	164.0	229.2	180.4	277.2	191.4	312.0	195.9	327.1
8	0.035	70.5	20.4	94.0	36.2	117.5	56.6	141.0	81.5	164.5	111.0	187.5	144.1	206.1	174.2	218.7	196.1	223.9	205.5
9	0.0133	79.3	7.7	105.8	13.8	132.2	21.5	158.7	31.0	185.1	42.2	210.9	54.7	231.9	66.2	246.0	74.5	251.9	78.1
9.6	0	84.6	0	112.8	0	141.0	0	169.2	0	197.4	0	225.0	0	247.4	0	262.4	0	268.7	0

Table 3 Calculated values of n and Q as a function of λ and V for the VIRYA-6.5 rotor

The calculated values for n and Q are plotted in figure 6. The optimum parabola which is going through the optimum points of the Q-n curves is also given in figure 6.

The measured unloaded Q_g-n generator curve has to be transformed to the rotor shaft to compare it with the rotor torque. It is assumed that the transmission efficiency η_{tr} = 0.95 for all working points. So it is valid that:

$$Q = Q_g * i / \eta_{tr} \quad \text{(Nm)} \qquad (29)$$

Substitution of i = 12.4 and η_{tr} = 0.95 in formula 3 gives that:

$$Q = 13.05 * Q_g \quad \text{(Nm)} \qquad (30)$$

The unloaded Q_g-n curve for star rectification is given in figure 1 of KD 200. The measuring points are copied from figure 1 of KD 200 and are given in table 4.

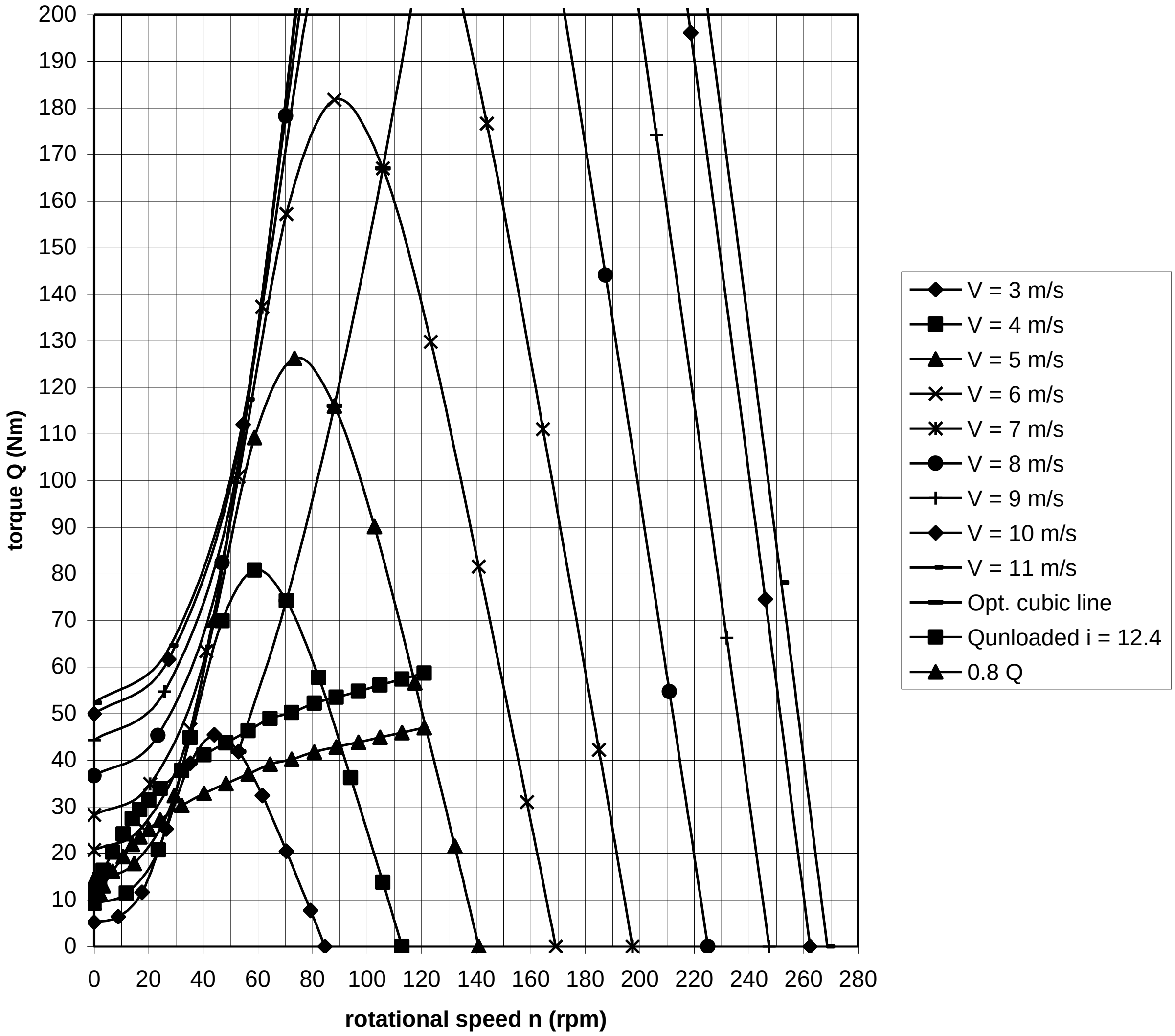

fig. 6 Q-n curves and optimum parabola for the VIRYA-6.5 rotor. Unloaded Q-n curve of the VIRYA-4.2 generator for star rectification for i = 12.4. Unloaded 0.8 Q-n curve of the VIRYA-4.2 generator with 120 mm stator for star rectification for i = 12.4.

n_g (rpm)	Q_g (Nm)	n (rpm) for i = 12.4	Q (Nm) for i / η_{tr} = 13.05	0.8 Q (Nm) for i / η_{tr} = 13.05
0	0.9	0	11.7	9.36
25	1.1	2.02	14.4	11.52
41	1.25	3.31	16.3	13.04
84	1.55	6.77	20.2	16.16
133	1.85	10.73	24.1	19.28
174	2.1	14.03	27.4	21.92
208	2.25	16.77	29.4	23.52
249	2.4	20.08	31.4	25.12
300	2.6	24.19	33.9	27.12
399	2.9	32.18	37.8	30.24
499	3.15	40.24	41.1	32.88
599	3.35	48.31	43.7	34.96
700	3.55	56.45	46.3	37.04
800	3.75	64.52	48.9	39.12
898	3.85	72.42	50.2	40.16
1000	4	80.65	52.2	41.76
1100	4.1	88.71	53.5	42.80
1201	4.2	96.85	54.8	43.84
1300	4.3	104.84	56.1	44.88
1400	4.4	112.90	57.4	45.92
1500	4.5	120.97	58.7	46.96

table 4 Measured unloaded generator torque Q_g as a function of n_g in star for the VIRYA-4.2 generator. Calculated values of n, Q and 0.8 Q for i = 12.4

The calculated values for n and Q are given in figure 6 as the unloaded Q-n curve of the generator for star rectification. In figure 6 it can be seen that the unloaded Q-n curve starts at a wind speed just below V = 5 m/s. So the rotor starts rotating at this wind speed. However, the unloaded Q-n curve touches about the P-n curve of the rotor for V = 7 m/s at a rotational speed of about 20 rpm. So this means that the rotor will only really start at a wind speed of 7 m/s and this is expected much too high.

The unloaded Q-n curve for a PM-generator with a stator length of 120 mm in stead of 150 mm will lie about a factor 0.8 lower. The value of 0.8 Q is given in the last column of table 4. The 0.8 Q curve is also given in figure 6. It can be seen that the curve for 0.8 Q is about touching the Q-n curve of the rotor for V = 6 m/s. This means that now the rotor will really start for a wind speed of about 6 m/s but this is also expected to be too high. So using a PM-generator derived from an asynchronous motor, so with iron in the coils, isn’t allowed in combination with a gear box with a high accelerating gear ratio and this idea is therefore cancelled.

8 References

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