

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° 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° 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:

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

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 $0.75 \lambda_d$ and $\lambda_{unloaded}$. Substitution of $C_t = 0.7$, $\delta = 30^\circ$, $\rho = 1.2 \text{ kg/m}^3$, $V = 11 \text{ m/s}$, $R = 3.25 \text{ m}$ and $B = 3$ in formula 1 gives $F_{t \delta bl} = 422 \text{ N}$.

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 $2/3 R = 2.167 \text{ m}$. 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 $r_1 = 2.28 \text{ m}$.

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 $r_2 = 0.075 \text{ m}$. At this edge we find a bending moment $M_{b t}$ caused by the thrust which is given by:

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

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

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:

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

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

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

(3) + (4) gives:

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

Substitution of $M = 930000 \text{ Nmm}$, $b = 120 \text{ mm}$ and $h = 15 \text{ mm}$ in formula 5 gives $\sigma_b = 207 \text{ N/mm}^2$. 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 $r_3 = 0.295 \text{ m} = 295 \text{ mm}$.

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

$$l = r_3 - r_2 \quad (\text{mm}) \quad (6)$$

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:

$$M = F * (r_1 - r_3) \quad (\text{Nmm}) \quad (7)$$

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):

$$\phi = l * (M + \frac{1}{2} Fl) / EI \quad (\text{rad}) \quad (8)$$

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

$$I = 1/12 bh^3 \quad (\text{mm}^4) \quad (9)$$

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

$$\phi = 12 * F * (r_3 - r_2) * \{(r_1 - r_3) + \frac{1}{2} (r_3 - r_2)\} / (E * bh^3) \quad (\text{rad}) \quad (10)$$

Substitution of $F = 422 \text{ N}$, $r_3 = 295 \text{ mm}$, $r_2 = 75 \text{ mm}$, $r_1 = 2280 \text{ mm}$, $E = 2.1 * 10^5 \text{ N/mm}^2$, $b = 120 \text{ mm}$ and $h = 15 \text{ mm}$ in formula 10 gives: $\phi = 0.02744 \text{ rad} = 1.57^\circ$. 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:

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

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 $0.7 * 44.5 * 300 = 9345 \text{ mm}^2 = 0.009345 \text{ m}^2$. The blade is made of hard wood with a density ρ_{pr} of about $\rho_{pr} = 0.6 * 10^3 \text{ kg/m}^3$. In figure 4 of KD 578 it can be seen that for high wind speeds, the rotor is running at about $\lambda = 4.5$. Substitution of $C_t = 0.7$, $\rho = 1.2 \text{ kg/m}^3$, $R = 3.25 \text{ m}$, $B = 3$, $A_{pr} = 0.009345 \text{ m}^2$, $\rho_{pr} = 0.6 * 10^3 \text{ kg/m}^3$ and $\lambda = 4.5$ in formula 11 gives: $\varepsilon = 4.69^\circ$. 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 as 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 $\varepsilon = 1.57^\circ$. A centrifugal moment M_c is working forwards and M_c is also proportional with ε . $M_c = 930$ Nm for $\varepsilon = 4.69^\circ$. 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.

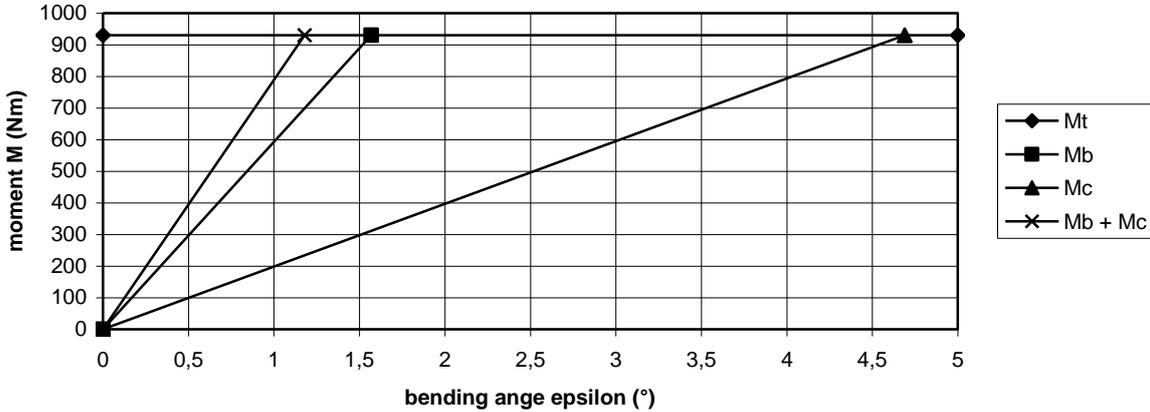


fig. 1 Path of M_t , M_b , M_c , and $M_b + M_c$ as a function of ε

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 $\varepsilon = 1.18^\circ$. 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/mm² resulting in a stress of about 156 N/mm². 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 $\sin\alpha$ 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:

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

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.

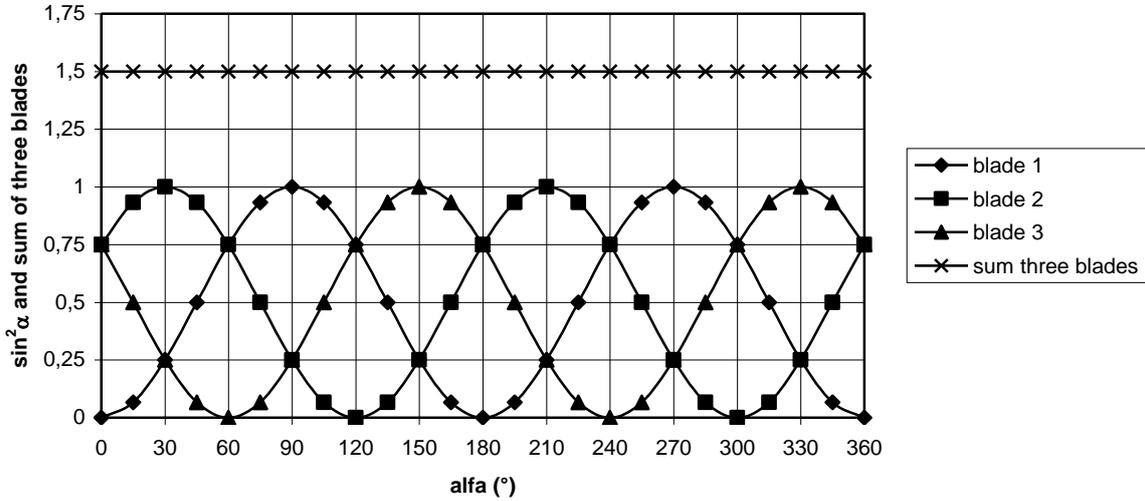


fig. 2 Path of $\sin^2\alpha$ and the sum of three blades

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:

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

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:

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

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:

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

For the chosen blade geometry it is calculated that $I_{\text{bl}} = 66 \text{ kgm}^2$. The maximum loaded rotational speed of the rotor can be read in figure 4 of KD 578 and it is found that $n_{\text{max}} = 128 \text{ rpm}$. This gives $\Omega_{\text{rot max}} = 13.4 \text{ rad/s}$ (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 $I_{bl} = 66 \text{ kgm}^2$, $\Omega_{rot \text{ max}} = 13.4 \text{ rad/s}$ en $\Omega_{head \text{ max}} = 0.2 \text{ rad/s}$ in formula 15 gives: $M_{gyr \text{ bl max}} = 212 \text{ Nm} = 212000 \text{ Nmm}$.

Substitution of $M = 212000 \text{ Nmm}$, $b = 150 \text{ mm}$ and $h = 15 \text{ mm}$ in formula 5 gives $\sigma_{b \text{ max}} = 38 \text{ N/mm}^2$. 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 where both moments are strengthening each other. This gives $\sigma_{b \text{ tot max}} = 194 \text{ N/mm}^2$. The minimum stress is $156 - 38 = 118 \text{ N/mm}^2$. 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 in the strip will be somewhat lower than the calculated value.

2.2 Bending stress in the spoke for a stopped rotor

It 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 $V = V_{rated} = 11 \text{ m/s}$ and $\delta = 30^\circ$. 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 $r_4 = 1.75 \text{ m}$ 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:

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

Substitution of $F_{t \delta \text{ bl}} = 422 \text{ N}$, $r_4 = 1.75 \text{ m}$ en $r_2 = 0.075 \text{ m}$ in formula 16 gives $M_{b \text{ t}} = 707 \text{ Nm} = 707000 \text{ Nmm}$. Substitution of $M = 707000 \text{ Nmm}$, $b = 120 \text{ mm}$ and $h = 15 \text{ mm}$ in formula 5 gives $\sigma_b = 157 \text{ N/mm}^2$. This is lower than the calculated stress for a rotating rotor. The load is not fluctuating and therefore it is surely 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 $1.57 * 157 / 207 = 1.19^\circ$. For a rotor radius of $R = 3.25 \text{ m}$ 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 $I_{rot} = 3 * 66 = 198 \text{ kgm}^2$. Substitution of $I_{rot} = 198 \text{ kgm}^2$, $\Omega_{rot} = 13.4 \text{ rad/s}$ and $\Omega_{head} = 0.2 \text{ rad/s}$ in formula 12 gives $M_{gyr} = 531 \text{ Nm} = 531000 \text{ Nmm}$.

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 $d = 48 \text{ mm}$ in formula 17 gives $W = 10857 \text{ mm}^3$. Substitution of $M = 531000 \text{ Nmm}$ and $W = 10857$ in formula 3 gives that $\sigma_b = 49 \text{ N/mm}^2$. 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 $1000 * 55 = 55000 \text{ Nmm}$. Substitution of this value and $W = 10857 \text{ mm}^3$ in formula 3 gives $\sigma_b = 5 \text{ N/mm}^2$ 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 $n = 123 \text{ rpm}$. Assume the wanted lifetime L is five years = $5 * 24 * 365 = 43800 \text{ hour}$. 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 $F = 3350 \text{ N}$ 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 $75 * 12.4 = 930 \text{ Nm}$. 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:

$$d_p = D_p - 2 * t \quad (\text{mm}) \quad (18)$$

W and I can be calculated by:

$$W = \pi/32 * (D_p^4 - d_p^4) / D_p \quad (\text{mm}^3) \quad (19)$$

$$I = \pi/64 * (D_p^4 - d_p^4) \quad (\text{mm}^4) \quad (20)$$

	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:

$$\begin{aligned} W_{3''} / W_{2''} &= 21907 / 8679 = 2.52 && \text{This is a factor 0.68 of 3.71} \\ I_{3''} / I_{2''} &= 973775 / 261669 = 3.72 && \text{This is a factor 0.65 of 5.74} \\ W_{5''} / W_{3''} &= 68796 / 21907 = 3.14 && \text{This is a factor 0.85 of 3.71} \\ I_{5''} / I_{3''} &= 4805412 / 973775 = 4.93 && \text{This is a factor 0.86 of 5.74} \end{aligned}$$

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.

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

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 $\alpha = 30^\circ$.

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 $\alpha = 30^\circ$ (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 $\alpha = 28^\circ$. This angle is 2° smaller than the angle $\alpha = 30^\circ$ for which the rotor is perpendicular to the wind. This means that the rotor makes a negative yaw angle $\delta = -2^\circ$ 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.

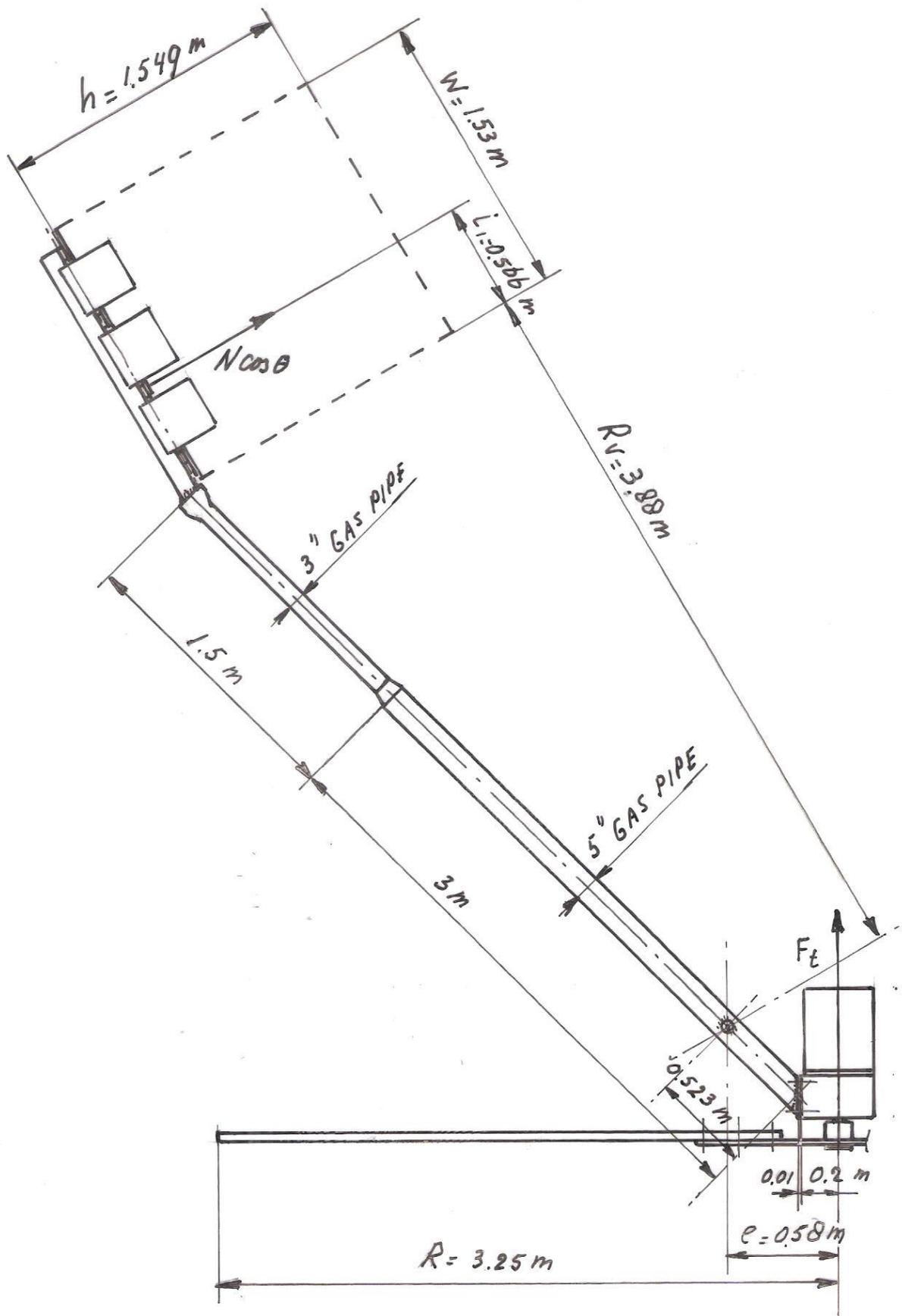


fig. 4 Top view head VIRYA-6.5 windmill

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.

$$F_{t\delta} = C_t * \cos^2\delta * \frac{1}{2}\rho V^2 * \pi R^2 \quad (\text{N}) \quad (22)$$

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 $\delta = 30^\circ$ 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 shear, 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:

$$F = C_d * \frac{1}{2}\rho V^2 * D_p * l \quad (\text{N}) \quad (23)$$

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 $C_d = 0.6$, $\rho = 1.2 \text{ kg/m}^3$, $V = 34$ m/s, $D_p = 0.1778$ m and $l = 5.4$ m in formula 23 gives that $F_u = 400$ N.

Substitution of $C_d = 0.6$, $\rho = 1.2 \text{ kg/m}^3$, $V = 31$ m/s, $D_p = 0.2445$ m and $l = 5.4$ m in formula 23 gives that $F_m = 457$ N.

Substitution of $C_d = 0.6$, $\rho = 1.2 \text{ kg/m}^3$, $V = 26$ 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:

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

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 $\sigma_U = 95$ N/mm^2 .

The bending moment M_M is given by:

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

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 $\sigma_M = 92$ N/mm^2 .

The bending moment M_L is given by:

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

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 $\sigma_M = 85$ N/mm^2 .

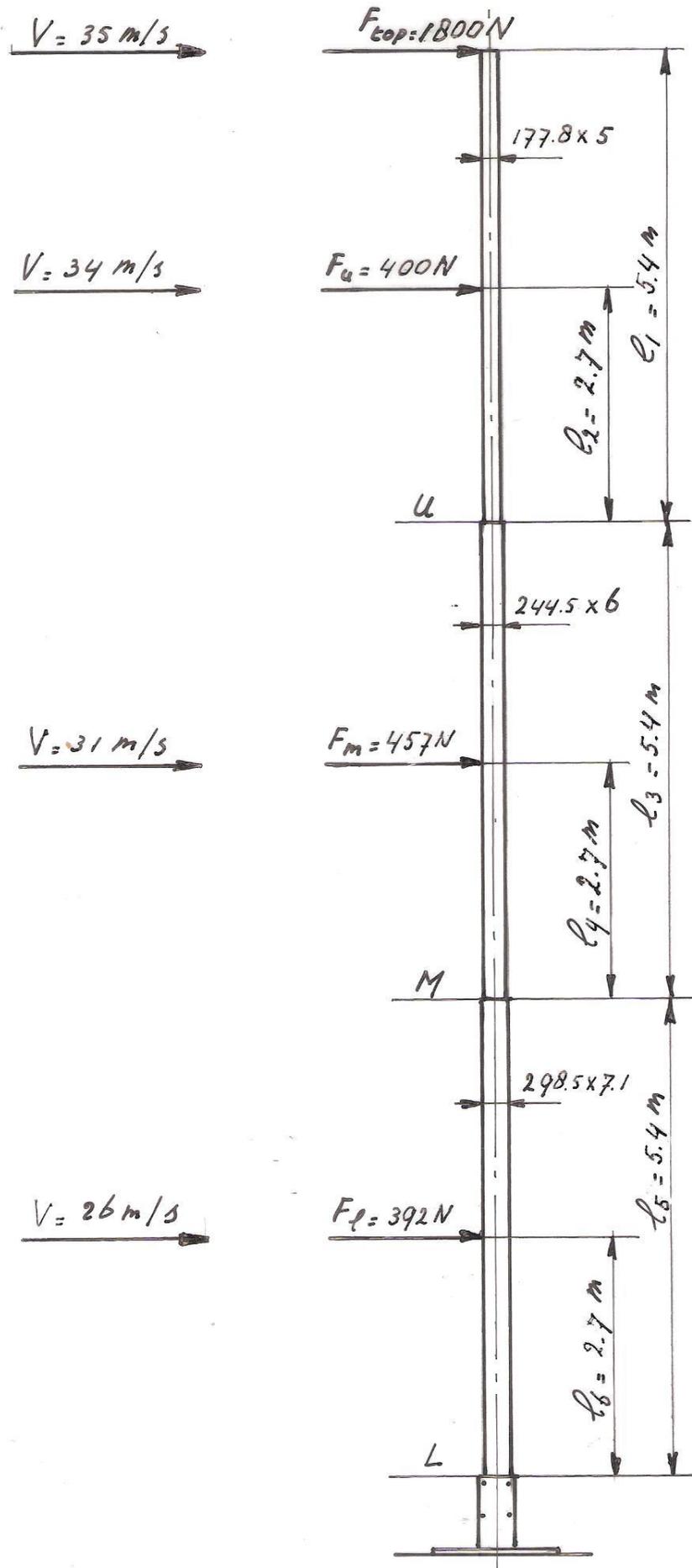


fig. 5 Forces acting on the tubular tower of the VIRYA-6.5

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 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 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 if the generator is provided with a low voltage winding. Another option is to use the original 230/400 V motor winding and use the 3-phase current directly to power the 3-phase asynchronous motor of a centrifugal pump. However, a PM-generator made from an asynchronous motor has iron in the coils which causes a certain sticking torque and this may result in a too high starting wind speed.

The use of the VIRYA-6.5 in combination with a direct drive 46-pole PM-generator is described in report KD 624 (ref. 6). The use of the VIRYA-6.5 in combination with a direct drive 50-pole PM-generator is described in report KD 650 (ref. 7). Both generators make use of the housing of an 8-pole asynchronous motor frame size 180 with a stator stamping of Kienle & Spiess. A generator of this frame size is rather heavy and the PM-armature will be expensive to manufacture. If a 4-pole PM-generator is used in combination with a gear box with a gear ratio of 12.4, the frequency is almost the same as for a 50-pole direct drive PM-generator as $12.4 * 4 = 49.6$. I think that the 4-pole PM-generator must be made from a 4 kW or a 3 kW motor but a PM-generator made from these motor sizes hasn't been tested by me. However, the characteristics can possibly be derived from other tested PM-generators.

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. 8). The VIRYA-4.2 generator makes use of a special 5.5 kW motor of manufacture ROTOR. 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 diameter of 107.94 mm and 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 $40 * 30 * 10 \text{ mm}$ are glued in each groove.

A 4-pole PM-generator with frame size 90 has been designed for the VIRYA-2.68 windmill and this generator has been measured for different loads for rotational speeds up to 1500 rpm. The measurements are given in report KD 78 (ref. 9). The VIRYA-2.68 generator makes use of a special 2.5 kW motor of manufacture ROTOR. The stator stamping of this motor has a length of 125 mm compared to a length of 100 mm for a normal 1.5 kW motor. The armature has a diameter of 82.02 mm and a length of 135 mm and has four inclined grooves. So the armature juts out 5 mm at each side of the stator. Two neodymium magnets size 67 * 20 * 10 mm are glued in each groove.

The PM-generator will be connected in star when it is connected to the asynchronous motor of a centrifugal pump. The starting behaviour is checked in the Q-n graph. So to check the starting behaviour, the measured unloaded Q-n curve for rectification in star has to be used as rectification in star gives the lowest sticking torque. The unloaded Q-n curve for a 4 kW, PM-generator will be derived from the measurements of the VIRYA-4.2 generator which uses a 5.5 kW housing. The unloaded Q-n curve for a 3 kW, PM-generator will be derived from the measurements of the VIRYA-2.68 generator which uses a 2.5 kW housing.

It must be possible to make a PM-generator which is made from a standard 4 kW motor of manufacture ROTOR 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 $120 / 150 = 0.8$. 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.

It must be possible to make a PM-generator which is made from a standard 3 kW motor frame size 100. A PM-generator made from a 2.2 kW motor with frame size 100 and using an Indian stator stamping is described in report KD 503 (ref. 10). For this PM-generator with a 3 kW housing, a stator stamping of Kienle & Spies (see website: www.kienle-spiess.de) will be chosen. The armature will have a diameter of 89.5 mm if the air gap in between stator and armature is 0.25 mm. The stator length will be 120 mm. So the armature volume lying within the stator becomes $\pi/4 * 89.5^2 * 120 = 754948 \text{ mm}^3$. The armature volume lying within the stator of the VIRYA-2.68 generator is $\pi/4 * 82.02^2 * 125 = 660449 \text{ mm}^3$. So the armature volume of the new generator is a factor $754948 / 660449 = 1.143$ larger than that of the VIRYA-2.68 generator. The torque level is proportional to the armature volume if the magnetic flux in the air gap in between stator and armature is the same. For the new generator, an armature with a length of 136 mm will be used. So the armature will jut out 8 mm at each side of the stator. Three neodymium magnets size 45 * 25 * 10 mm will be used in each groove and so the magnet volume increases by a factor $3 * 45 * 25 / (2 * 67 * 20) = 1.259$. The magnetic flux in the air gap will therefore be larger than for the VIRYA-2.68 generator. It is assumed that the torque level will therefore increase by a factor 1.2. It might be possible to use the original motor shaft if it is provided with a stainless steel bush to prevent magnetic short-circuit in between the armature poles.

The next point which has to be checked is the increase of the starting wind speed due to the sticking torque of the generator. 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 $C_q \cdot \lambda$ curve in stead of the $C_p \cdot \lambda$ 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:

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

Substitution of $\rho = 1.2$ kg / m³ and $R = 3.25$ m in formula 7.7 of KD 35 gives:

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

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.

λ	C_q	V = 3 m/s $\delta = 0^\circ$		V = 4 m/s $\delta = 0^\circ$		V = 5 m/s $\delta = 0^\circ$		V = 6 m/s $\delta = 0^\circ$		V = 7 m/s $\delta = 0^\circ$		V = 8 m/s $\delta = 4.5^\circ$		V = 9 m/s $\delta = 13^\circ$		V = 10 m/s $\delta = 21.5^\circ$		V = 11 m/s $\delta = 30^\circ$	
		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_g generator curve has to be transformed to the rotor shaft to compare it with the rotor torque. It is assumed that the transmission efficiency $\eta_{tr} = 0.95$ for all working points. So it is valid that:

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

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

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

The relation in between the rotational speed of the generator shaft n_g and the rotational speed of the rotor shaft n is given by:

$$n = n_g / 12.4 \quad (\text{rpm}) \quad (31)$$

The unloaded Q_g - n curve for star rectification of the VIRYA-4.2 generator is given in figure 1 of KD 200. The measuring points are given as Q_g and n_g in table 4. The measuring points multiplied by a factor 0.8 are given as $Q_{g0.8}$ in table 4. These are the estimated values for the new generator using a 4 kW housing.

The unloaded Q_g - n curve for star rectification of the VIRYA-2.68 generator is given in figure 2 of KD 78. The measuring points are given as Q_g and n_g in table 5. The measuring points multiplied by a factor 1.2 are given as $Q_{g1.2}$ in table 5. These are the estimated values for the new generator using a 3 kW housing.

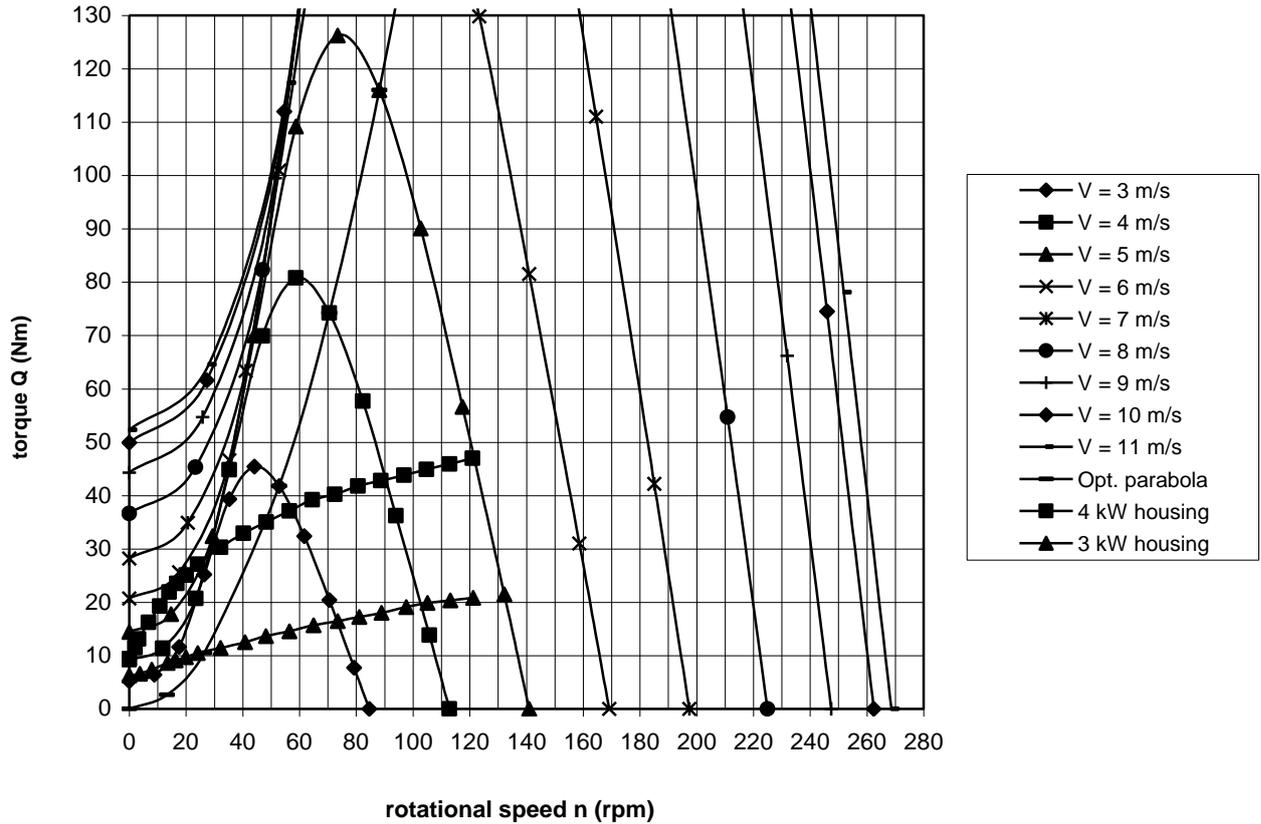


fig. 6 Q-n curves and optimum parabola for the VIRYA-6.5 rotor. Estimated unloaded $Q_{0.8-n}$ curve of the 4 kW PM-generator for $i = 12.4$. Estimated unloaded $Q_{1.2-n}$ curve of the 3 kW PM-generator for $i = 12.4$.

n_g (rpm)	Q_g (Nm)	$Q_{g0.8}$ (Nm)	n (rpm) for $i = 12.4$	$Q_{0.8}$ (Nm) for $i / \eta_{tr} = 13.05$
0	0.9	0.72	0	9.4
25	1.1	0.88	2.02	11.5
41	1.25	1.00	3.31	13.1
84	1.55	1.24	6.77	16.2
133	1.85	1.48	10.73	19.3
174	2.1	1.68	14.03	21.9
208	2.25	1.80	16.77	23.5
249	2.4	1.92	20.08	25.1
300	2.6	2.08	24.19	27.1
399	2.9	2.32	32.18	30.3
499	3.15	2.52	40.24	32.9
599	3.35	2.68	48.31	35.0
700	3.55	2.84	56.45	37.1
800	3.75	3.00	64.52	39.2
898	3.85	3.08	72.42	40.2
1000	4	3.20	80.65	41.8
1100	4.1	3.28	88.71	42.8
1201	4.2	3.36	96.85	43.8
1300	4.3	3.44	104.84	44.9
1400	4.4	3.52	112.90	45.9
1500	4.5	3.60	120.97	47.0

table 4 Measured unloaded generator torque Q_g as a function of n_g for the VIRYA-4.2 generator. Estimated unloaded generator torque $Q_{g0.8}$ as a function of n_g in star for a PM-generator with a 4 kW housing. Calculated values of n , $Q_{0.8}$ for $i = 12.4$

n_g (rpm)	Q_g (Nm)	$Q_{g1.2}$ (Nm)	n (rpm) for $i = 12.4$	$Q_{1.2}$ (Nm) for $i / \eta_r = 13.05$
0	0.4	0.48	0	6.26
46	0.42	0.504	3.71	6.58
99	0.47	0.564	7.98	7.36
167	0.55	0.66	13.47	8.61
203	0.58	0.696	16.37	9.08
248	0.62	0.744	20.00	9.71
299	0.67	0.804	24.11	10.49
400	0.73	0.876	32.26	11.43
506	0.8	0.96	40.81	12.53
598	0.87	1.044	48.23	13.62
700	0.93	1.116	56.45	14.56
806	1.0	1.2	65.00	15.66
911	1.05	1.26	73.47	16.44
1005	1.1	1.32	81.05	17.23
1103	1.15	1.38	88.95	18.01
1211	1.22	1.464	97.66	19.11
1303	1.27	1.524	105.08	19.89
1404	1.3	1.56	113.23	20.36
1503	1.33	1.596	121.21	20.83

table 4 Measured unloaded generator torque Q_g as a function of n_g for the VIRYA-2.68 generator. Estimated unloaded generator torque $Q_{g1.2}$ as a function of n_g in star for a PM-generator with a 3 kW housing. Calculated values of n , $Q_{1.2}$ for $i = 12.4$

The calculated values for n and $Q_{0.8}$ are given in figure 6 as the unloaded $Q_{0.8}$ - n curve of the generator with a 4 kW housing for star rectification. The calculated values for n and $Q_{1.2}$ are given in figure 6 as the unloaded $Q_{1.2}$ - n curve of the generator with a 3 kW housing for star rectification. In figure 6 it can be seen that the $Q_{0.8}$ - n curve for the 4 kW housing is rising much faster than the $Q_{1.2}$ - n curve for the 3 kW housing. The measurements performed for the VIRYA-4.2 and the VIRYA-2.68 are very accurate and I don't know the reason why the curves differ that much.

In figure 6 it can be seen that the unloaded $Q_{0.8}$ - n curve for a 4 kW housing starts at a wind speed of about $V = 4$ m/s. So the rotor starts rotating at this wind speed. However, the unloaded $Q_{0.8}$ - n curve touches about the P- n curve of the rotor for $V = 6$ m/s at a rotational speed of about 16 rpm. So this means that the rotor will only really start at a wind speed of 6 m/s and this is expected to be much too high.

In figure 6 it can be seen that the unloaded $Q_{1.2}$ - n curve for a 3 kW housing starts at a wind speed of about $V = 3$ m/s. So the rotor starts rotating at this wind speed. However, the unloaded $Q_{1.2}$ - n curve touches about the P- n curve of the rotor for $V = 3.2$ m/s (not drawn) at a rotational speed of about 10 rpm. So this means that the rotor will really start at a wind speed of about 3.2 m/s and this is certainly acceptable. So even if the real curve of the generator with a 3 kW housing would rise faster than the estimated curve, the starting behaviour will still be acceptable.

So assume that a PM-generator with a 3 kW housing is chosen. The question now is, if this generator is strong enough to load the windmill rotor sufficiently at high wind speeds. In figure 8 of KD 78 it can be seen that the maximum torque level of the VIRYA-2.68 generator is about 29 Nm. The maximum torque level of the new PM-generator with a 3 kW housing will be about a factor 1.2 higher so about 34.8 Nm. This is the torque at the generator shaft. Using formula 30 it can be calculated that the maximum torque level at the rotor shaft will be about $34.8 * 13.05 = 454$ Nm.

The rotor torque Q is given in table 3 depending on the tip speed ratio λ and the wind speed V . Next it is assumed that the rotor is loaded by the generator such that it runs at a tip speed ratio $\lambda = 6$. In table 3 it can be read that this gives a torque $Q = 421$ Nm at a rotational speed $n = 167.9$ rpm.

Using formula 31 it can be calculated that the rotational speed of the generator $n_g = 167.9 * 12.4 = 2082$ rpm. The frequency of a 4-pole motor is 50 Hz if it runs at a rotational speed of 1500 rpm so the frequency is $50 * 2082 / 1500 = 69.4$ Hz if it runs at 2082 rpm. So the peak torque which the generator can supply is just a bit higher than the rotor torque at $V = 11$ m/s and $\lambda = 6$ and the generator frequency is already rather high. I think that the generator is too light for this high rotor torque. This problem can be solved by reducing the rated wind speed of the rotor by taking a lighter vane blade.

In chapter 5 it was assumed that the vane blade is made of a okume plywood with size $1530 * 1559$ mm. In chapter 5 of KD 578 it is chosen that the vane blade has a thickness of 15 mm and it is assumed that this results in a rated wind speed of 11 m/s. The rated wind speed can be reduced by taking a lighter vane blade but reducing the thickness from 15 mm to for instance 12 mm will make the vane blade too flexible and therefore too sensible for flutter at high wind speeds. This problem can be solved by making a sandwich construction. The thinnest available standard sheet size $3100 * 1530$ mm has a thickness of 5 mm. Assume that two sheets size $1530 * 1559 * 5$ mm are separated by 10 mm thick wooden strips. So the total thickness becomes 20 mm which makes the sheet rather stiff. The 10 mm wooden strips makes that the total weight is about the same as that of a 15 mm massive sheet. So the weight is reduced by a factor $11/15$. As the force on the blade is proportional to V^2 , this means that the rated wind speed is reduced by a factor $(11/15)^{1/2} = 0.856$ and it becomes about $0.856 * 11 = 9.42$ m/s. The torque is also proportional to V^2 and this means that the maximum torque of the rotor is reduced by a factor $11/15 = 0.733$. For this maximum torque level a PM-generator with a 3 kW housing will certainly be strong enough.

A 3 kW motor housing has a smaller flange than a 4 kW motor housing. So now one has to select a gear box which has a smaller flange too. The code number of this gear box meant for a 3 kW motor is: Rossi MR 2l 100-100L 4, $i = 12.4$. It should be investigated if the stator stamping of the housing of the asynchronous motor which is normally supplied by Rossi has the correct dimensions to be used for the PM-generator. If not, a motor with a stator stamping of Kienle & Spiess should be used.

In chapter 2 of KD 650 it is assumed that the maximum rotational speed and so the maximum frequency is limited by adding a voltage controller and dump load which limits the maximum AC voltage at 240 V corresponding to a DC voltage of 560 V. If a prototype of the new generator is available, it has to be measured how the real generator torque Q_g varies as a function of the rotational speed n_g for this DC voltage of 560 V. This measured Q_g - n_g curve then has to be transformed to the Q - n curve on the rotor shaft using formula 30 and 31. The Q - n curve of the generator is then drawn in the Q - n graph of the rotor to check if the maximum rotational speed is really limited enough. One should also check if the generated open AC phase voltage at 1500 rpm is high enough for the standard 230/400 V winding. If not, it may be required to use a special winding with more turns per coil.

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