

**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  $120^\circ$ . 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  $30^\circ$  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  $\lambda$  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  $M_{bt}$  caused by the thrust which is given by:

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

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  $M_{bt} = 930 \text{ Nm} = 930000 \text{ Nmm}$ .

For the stress we use the unit  $\text{N/mm}^2$  so the bending moment has to be given in  $\text{Nmm}$ . The bending stress  $\sigma_b$  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  $\phi$  (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  $\varepsilon$  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  $\text{m}^2$ ) and  $\rho_{pr}$  is the density of the used airfoil material (in  $\text{kg/m}^3$ ). 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  $\rho_{pr}$  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  $1.57^\circ$  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  $1.57^\circ$ .

The real bending angle  $\varepsilon$  is determined as follows. A thrust moment  $M_t = 930$  Nm is working backwards and  $M_t$  is independent of  $\varepsilon$  for small values of  $\varepsilon$ . A bending moment  $M_b$  is working forwards and  $M_b$  is proportional with  $\varepsilon$ .  $M_b = 930$  Nm for  $\varepsilon = 1.57^\circ$ . A centrifugal moment  $M_c$  is working forwards and  $M_c$  is also proportional with  $\varepsilon$ .  $M_c = 930$  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.

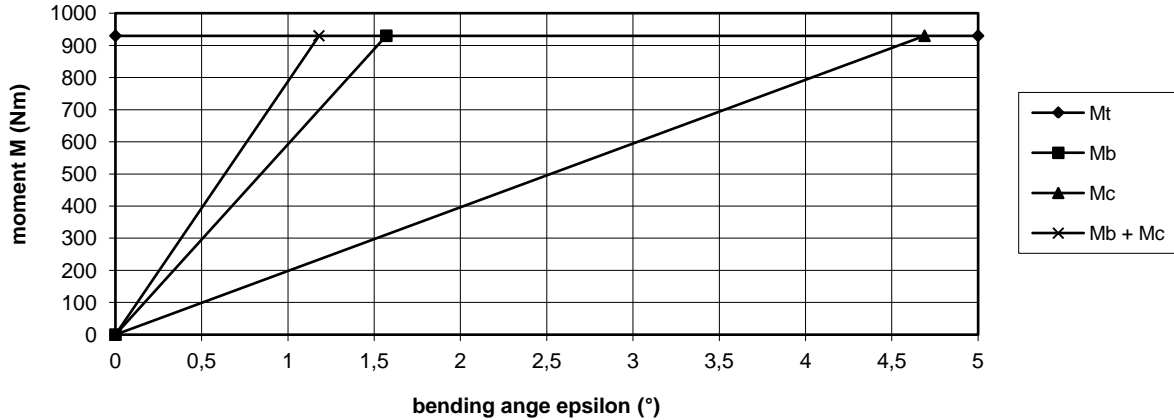


fig. 1 Path of  $M_t$ ,  $M_b$ ,  $M_c$ , and  $M_b + M_c$  as a function of  $\varepsilon$

The point of intersection of the line of  $M_t$  with the line of  $M_b + M_c$  gives the final angle  $\varepsilon$ . In figure 1 it can be seen that  $\varepsilon = 1.18^\circ$ . This is a factor 0.75 of the calculated angle of  $1.57^\circ$ . 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<sup>2</sup> resulting in a stress of about  $156$  N/mm<sup>2</sup>. 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.  $\alpha$  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  $I_{\text{rot}}$  is the mass moment of inertia of the whole rotor around the axis of rotation,  $\Omega_{\text{rot}}$  is the angular velocity of the rotor and  $\Omega_{\text{head}}$  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  $120^\circ$  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.

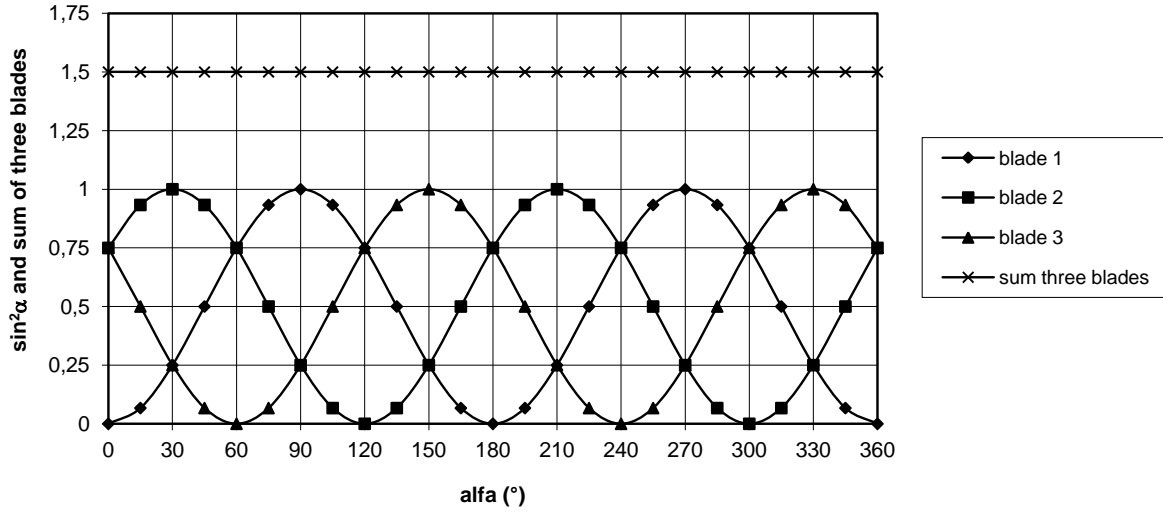


fig. 2 Path of  $\sin^2\alpha$  and the sum of three blades

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  $I_{\text{rot}}$  is three times the moment of inertia of one blade  $I_{\text{bl}}$ . 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  $156 \text{ N/mm}^2$  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  $300 \text{ N/mm}^2$ . However, this is for a pulling stress. The deformation limit for a bending stress is higher and it is expected that it is about  $400 \text{ N/mm}^2$ . 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  $30^\circ$  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:

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

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  $M_{bt} = 707 \text{ Nm} = 707000 \text{ Nmm}$ . 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  $1.57^\circ$  for a stress of  $207 \text{ N/mm}^2$ . So for a stress of  $157 \text{ N/mm}^2$  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  $W = 10857 \text{ mm}^3$ . The moment of resistance of a spoke can be calculated with formula 4 and it is found that  $W = 4500 \text{ mm}^3$ . So a spoke is a factor  $4500 / 10857 = 0.414$  weaker than the shaft. 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.

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 braking torque at the slow gear box shaft is  $75 * 12.4 = 930 \text{ Nm}$ . The rotor torque is maximal for a 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 <sup>3</sup> )	$I$ (mm <sup>4</sup> )
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 213. 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  $C_n$ - $\alpha$  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  $\alpha$  in between the wind direction and the vane blade is  $30^\circ$ . In figure 6 of KD 213, 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  $\alpha$  but is about  $0.37 * w$  for  $\alpha = 30^\circ$  (see figure 7 KD 213 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 213 (or figure 2 of KD 551) it can be seen that  $C_n = 1.28$  belongs about to  $\alpha = 28^\circ$ . This angle is  $2^\circ$  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  $15^\circ$  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  $3^\circ$  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  $45^\circ$  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  $5^\circ$  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.

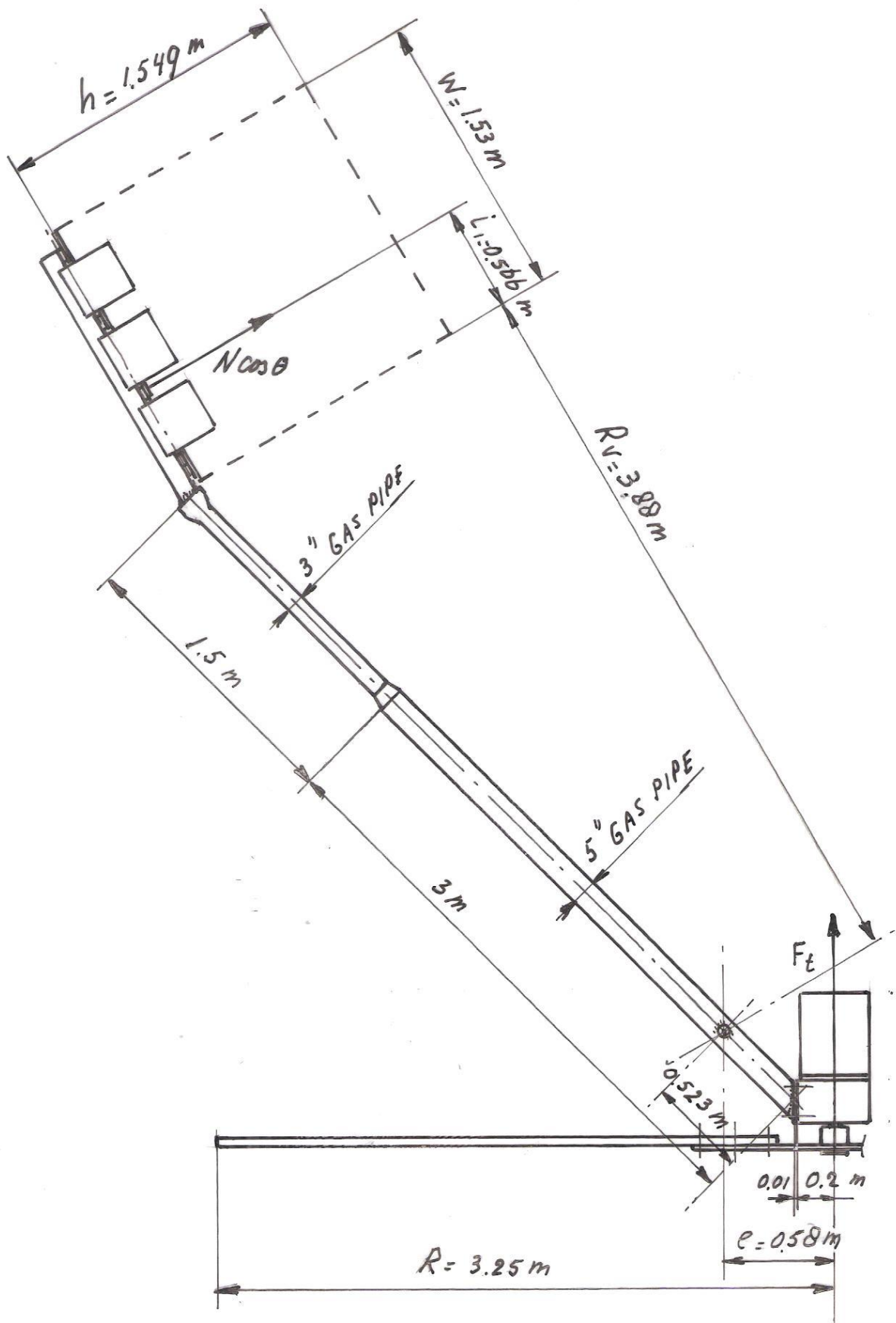


fig. 4 Top view head VIRYA-6.5 windmill

## 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  $\delta$  at  $V = 11$  m/s is  $30^\circ$  (see KD 578 figure 3). The thrust at a yaw angle  $\delta$ ,  $F_{t\delta}$  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.  $\rho$  is the air density which is about  $1.2 \text{ kg/m}^3$  for air of  $20^\circ \text{ C}$  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  $\sigma$  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.  $\sigma$  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  $\sigma_M = 85$   $N/mm^2$ .



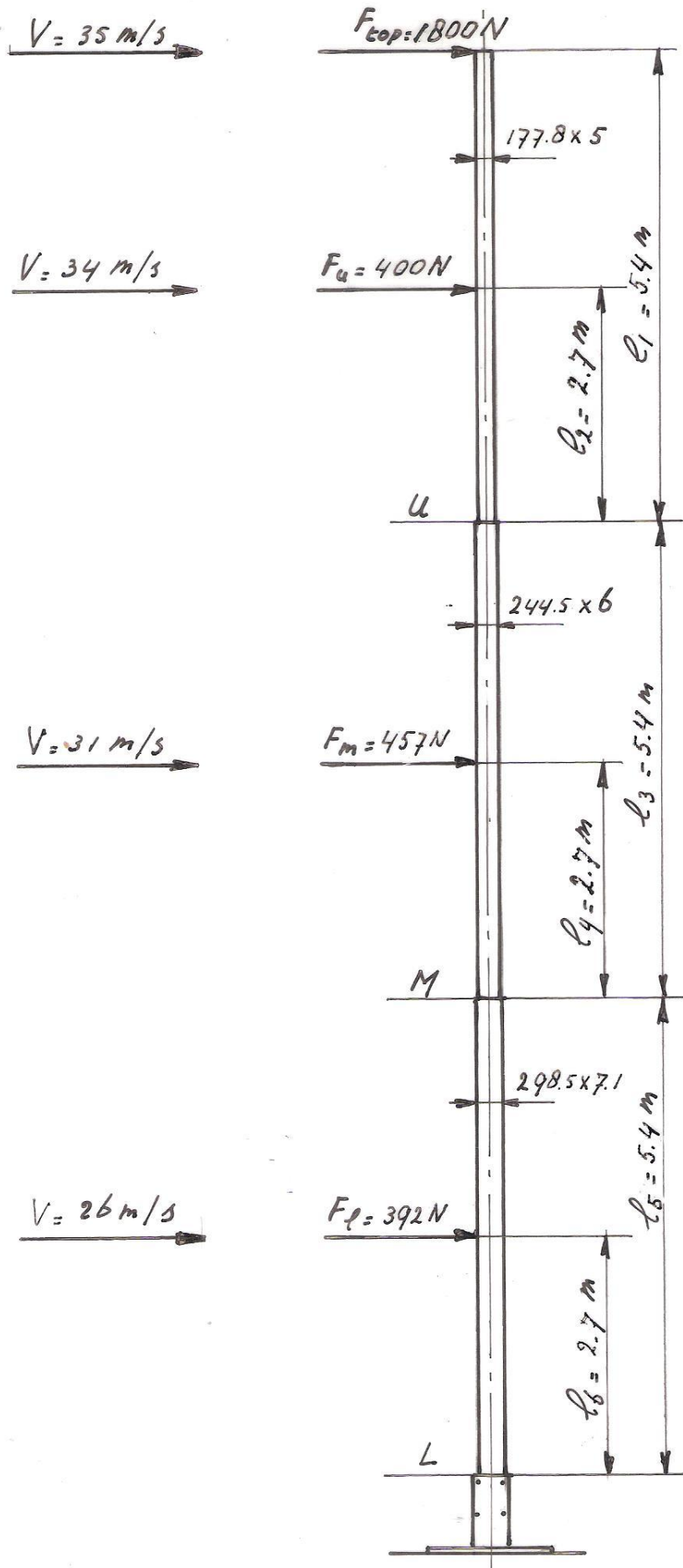


fig. 5 Forces acting on the tubular tower of the VIRYA-6.5

So the bending stress in cross sections U and M is about the same and about  $93 \text{ N/mm}^2$ . The bending stress in cross section L is somewhat lower and  $85 \text{ N/mm}^2$ . 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  $180 \text{ N/mm}^2$  if the load is not a fatigue load. However, the allowable bending stress is higher than the allowable pulling stress and is about  $240 \text{ N/mm}^2$ . 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 type PMG-I-620-10kW-250R**

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 use the VIRYA-6.5 in combination with a PM-generator. 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). These generators have a high frequency and are meant to be coupled to the asynchronous motor of a centrifugal pump. 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.

For grid connection, a high frequency isn't important and it might be possible to use a direct drive axial flux generator. Axial flux means that the magnetic flux which is flowing through the coils is in parallel to the generator axis. There is no iron in the coils and so the sticking torque is only determined by the friction of the bearings and the seal on the shaft. As there is no iron in the coils, there are no magnetic losses and the peak efficiency is rather high. Such generators are supplied by different Chinese suppliers like Hefei Top Grand, Xinda Green Energy, Hiestmotor and Qiangsheng Magnets. I have chosen Hefei Top Grand, website: [www.china-topgrand.com](http://www.china-topgrand.com) because they gave the clearest answers on my questions. I have bought and tested a much smaller generator type TGET165-0.15kW-500R at this company and they keep their promises. Measurements for this generator and experiments with a small wind turbine are given in report KD 595 (ref. 8).

Hefei top grand supplies PM-generators with an outer rotor indicated by TGET and with an inner rotor indicated with PMG. For the type TGET, the whole generator housing is rotating and the cables are coming out of the hollow generator shaft. For the type PMG, only the shaft is rotating and the cables are coming out of the back bearing cover. It appears that no generator with type TGET can be found which is strong enough fore the VIRYA-6.5. But for a type PMG a generator is found which seems strong enough.

For the VIRYA-6.5, it is chosen to the generator with type PMG620-I-10KW-250R. 620 refers to about the housing diameter in mm, the real diameter is 625 mm. 10KW refers to the nominal electric generator power 250R refers to the nominal rotational speed. A data sheet can be found on the website of the supplier following the path: [www.china-topgrand.com](http://www.china-topgrand.com) – product – Permanent Magnet Generator Inner Rotor – page 4.

In the data sheet it is specified that the efficiency is higher than 85 % which seems realistic if the load is a constant resistance. It is also specified that the electrical power is 10000 W, so 10 kW at  $n = 250$  rpm. Next it is assumed that the efficiency is 85 % or 0.85 (-). So the required mechanical power is  $10000 / 0.85 = 11765$  W. The generator has a mass of 140 kg which seems acceptable for the VIRYA-6.5.

The rated loaded voltage at  $n = 250$  rpm is specified as 450 VAC. So no DC voltage is specified but the loaded DC voltage can be calculated. This generator has a 3-phase winding with an internal star point and three phase wires are coming out of the generator housing. The given voltage is the voltage in between two of the three phases and not the phase voltage  $U_f$ , which is the voltage in between the star point and one of the phases.  $U_f$  is a factor  $\sqrt{3}$  lower, so 259.8 VAC. A large 3-phase rectifier (not included) must be used to get a DC current which is needed for the inverter. Rectification of a 3-phase current is explained in report KD 340 (ref. 9). However, it might be that the rectifier is included in the inverter and in this case the three phase wires are directly connected to the inverter. To stop the rotor, a 3-phase short-circuit switch has to be mounted at the tower foot. The switch must be mounted before the rectifier and as close as possible to the generator to prevent a voltage drop over the lines in between the generator and the switch.

The rated line current  $I$  is specified as  $I = 12.83$  A at  $n = 250$  rpm. So the rated power generated by one phase is  $U_f * I = 259.8 * 12.83 = 3333$  W. So the rated power generated by three phases is  $3 * 3333 = 9999$  W. This matches well with the given rated power of 10 kW.

The sticking torque of the generator is very low without an oil seal and is only caused by the friction of the bearings. It is specified that this torque is less than 0.5 Nm. An oil seal is needed if the axis is horizontal. The sticking torque will be much higher if an oil seal is mounted and if an auxiliary shaft is used but it is expected that it is low enough for the VIRYA-6.5 rotor (see calculation of the starting wind speed in chapter 4 of KD 578).

The generator has a shaft with a diameter of 80 mm which is certainly large enough for a windmill rotor with a diameter of 6.5 m. The length of the cylindrical part of the shaft is 210 mm which is rather long. The generator shaft has a 22 mm wide key groove. The front side of the shaft is provided with thread M72 \* 2. A composite drawing of the hub and the generator bracket of the head have still to be made.

The generator housing has a collar with a diameter of 200 mm at the front and at the back side. In the photo of the generator it can be seen that eight threaded holes are made in each collar but the diameter of the threaded holes and the diameter of the pitch circle aren't specified. As these threaded holes are used to connect the generator to the generator bracket of the head frame, the measures should have been given. The measures have still to be gained from the supplier to design the head frame.

The generator characteristics are given in point 6 of the data sheet. The  $P_{el}$ - $n$  and the loaded  $U$ - $n$  curves are given. The curves show measuring points but the measuring points aren't given in a table. The  $U$ - $n$  curve is about a straight line through the origin and the  $P_{el}$ - $n$  curve is about a parabola. This is an indication that the load is a fixed resistance for the whole range of measurements. I have performed this kind of measurements on a radial flux PM-generator made from an asynchronous motor. These measurements are given in chapter 7 and 9 of report KD 78 (ref. 10). These measurements show that the  $P_{mech}$ - $n$  and  $P_{el}$ - $n$  curves are about parabolas if the resistance isn't very low, that the  $U$ - $n$  and  $Q$ - $n$  curves are about straight lines through the origin and that the efficiency is about constant for a certain resistance. So these curves are estimated for the given generator of Hefei Top Grand from the given rated values at  $n = 250$  rpm.

The given rated voltage is the alternating voltage  $U_{AC}$  in between two of the three phases. For an inverter, the winding must be rectified. The rectified DC voltage  $U_{DC}$  is a factor  $0.955 * \sqrt{2} = 1.3506$  higher than  $U_{AC}$  (if the voltage drop of the rectifier diodes is neglected). The unloaded or open voltage  $U_{open}$  is also not specified. For a smaller generator type TGET320-1KW-350R, it has been found in chapter 3 of report KD 705 (ref. 11) that the ratio  $U_{open} / U_{DC}$  is about  $68 / 56 = 1.2143$ . It is assumed that this ratio is also valid for the generator type PMG620-I-10KW-250R. So for the loaded DC voltage  $U_{DC}$  at  $n = 250$  rpm it is valid that  $U_{DC} = 1.3506 * 450 = 608$  V DC. For the open DC voltage  $U_{open}$  at  $n = 250$  rpm it is valid that  $U_{open} = 1.2143 * 1.3506 * 450 = 739$  V DC. The calculated values are given in the bottom line of table 3.

n (rpm)	$U_{AC}$ (V)	$U_{DC}$ (V)	$U_{open}$ (V)	$P_{el}$ (W)	$\eta_{gen}$ (-)	$P_{mech}$ (W)	Q (Nm)	$P_{heat}$ (W)
0	0	0	0	0	-	0	0	0
25	45	60.8	73.8	100	0.85	118	44.9	18
50	90	121.6	147.6	400	0.85	471	89.8	71
75	135	182.4	221.4	900	0.85	1059	134.7	159
100	180	243.2	295.2	1600	0.85	1882	179.6	282
125	225	304.0	369.0	2500	0.85	2941	224.5	441
150	270	364.8	442.8	3600	0.85	4235	269.4	635
175	315	425.6	516.6	4900	0.85	5765	314.3	865
200	360	486.4	590.4	6400	0.85	7530	359.2	1130
225	405	547.2	664.2	8100	0.85	9530	404.1	1430
250	450	608	738	10000	0.85	11765	449	1765

table 3  $U_{AC}$ ,  $U_{DC}$ ,  $U_{open}$ ,  $P_{el}$ ,  $\eta_{gen}$ ,  $P_{mech}$ , Q and  $P_{heat}$  as a function of n

No rated torque Q is given for the generator. However, it is specified at point 5 of the data sheet that the generator efficiency  $\eta_{gen}$  is at least 85 %. In figure 33 of KD 78 (ref. 10) it can be seen that the efficiency for a resistance load is about constant for every rotational speed and that it is high if the load resistance isn't low. It is easy to give the efficiency as a factor of 1 and it is assumed that  $\eta_{gen} = 0.85$  for all rotational speeds. As the generator has no iron in the coils, the heat losses  $P_{heat}$  are only caused by the copper losses in the winding. The  $P_{mech}$ -n, the  $P_{heat}$ -n and the Q-n curves of the generator can be derived by the formulas:

$$P_{mech} = P_{el} / \eta_{gen} \quad (\text{W}) \quad (27)$$

$$P_{heat} = P_{mech} - P_{el} \quad (\text{W}) \quad (28)$$

$$Q = 30 P_{mech} / (\pi * n) \quad (\text{Nm}) \quad (29)$$

First the values of  $P_{mech}$ ,  $P_{heat}$  and Q are determined for  $n = 250$  rpm. Substitution of  $P_{el} = 10000$  W and  $\eta_{gen} = 0.85$  in formula 27 gives that  $P_{mech} = 11765$  W. Substitution of  $P_{mech} = 11765$  W and  $P_{el} = 10000$  W in formula 28 gives that the heat loss  $P_{heat} = 1765$  W. Substitution of  $P_{mech} = 11765$  W and  $n = 250$  rpm in formula 12 gives that  $Q = 449$  m. These values are also given in the bottom line of table 3.

The values for other rotational speeds are now calculated assuming that the U-n and Q-n curves are straight lines through the origin and that the P-n curves are parabolas. The wanted curves can now be derived from table 3. The  $U_{AC}$ -n, the  $U_{DC}$ -n and the  $U_{open}$ -n curves are given in figure 6.

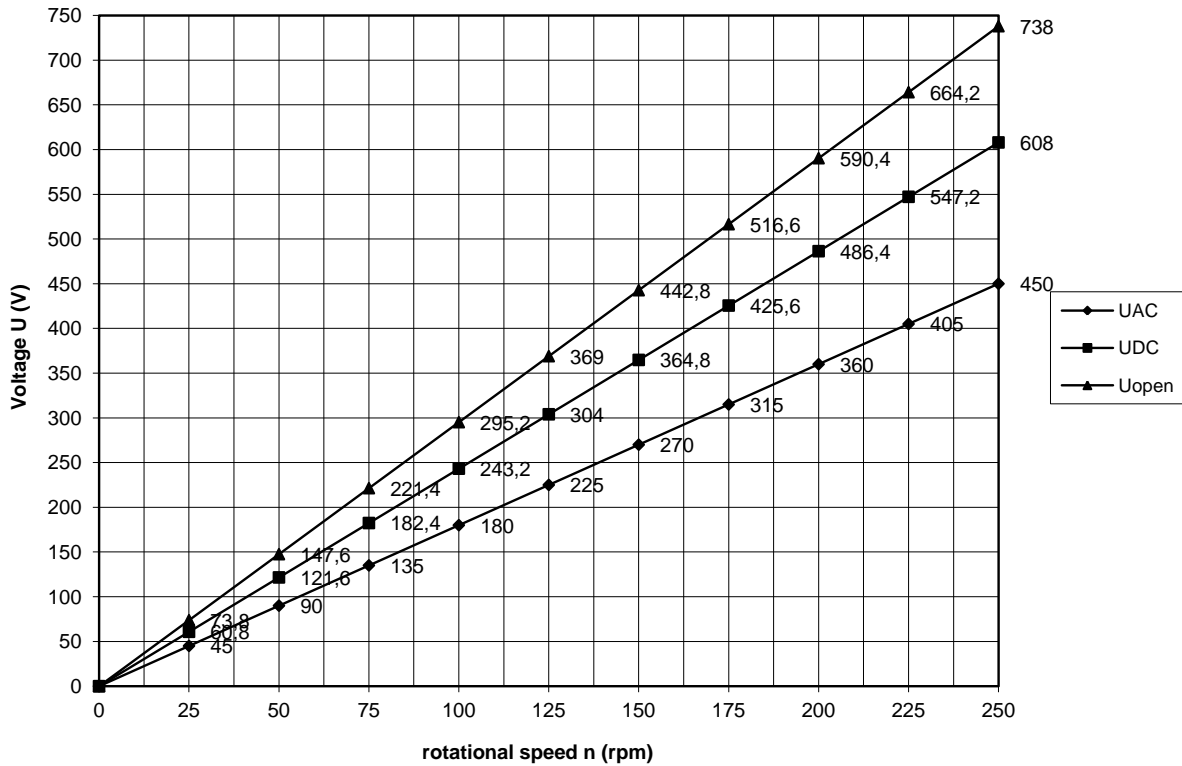


fig. 6  $U_{AC}$ ,  $U_{DC}$  and  $U_{open}$  as a function of  $n$  for a resistance load

The  $Q$ - $n$  curve is given in figure 7. Figure 7 also contains the curve for  $U = 608$  V DC and for short-circuit in star.

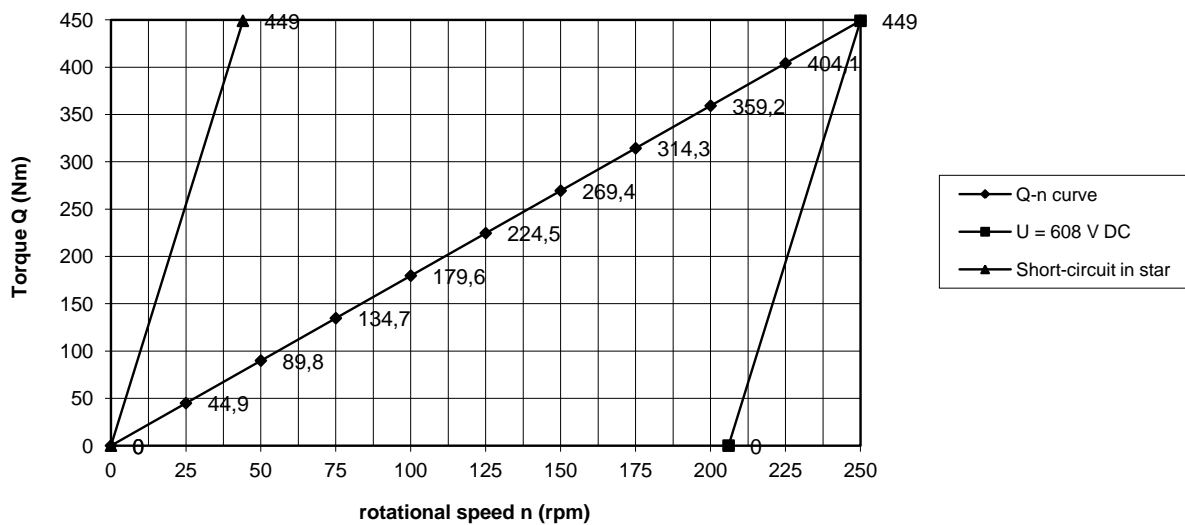


fig. 7 Loaded torque  $Q$  as a function of  $n$  for a resistance load,  $Q$ - $n$  curves for 608 V DC and for short-circuit in star

The P-n curves of the rotor are given in figure 4 of KD 578. This figure is copied as figure 8 but the generator curves for  $i = 12.4$  and  $i = 15.3$  are removed. The  $P_{\text{mech-n}}$  and  $P_{\text{el-n}}$  curves for a resistance load which can be derived from table 3 are also given in figure 8.

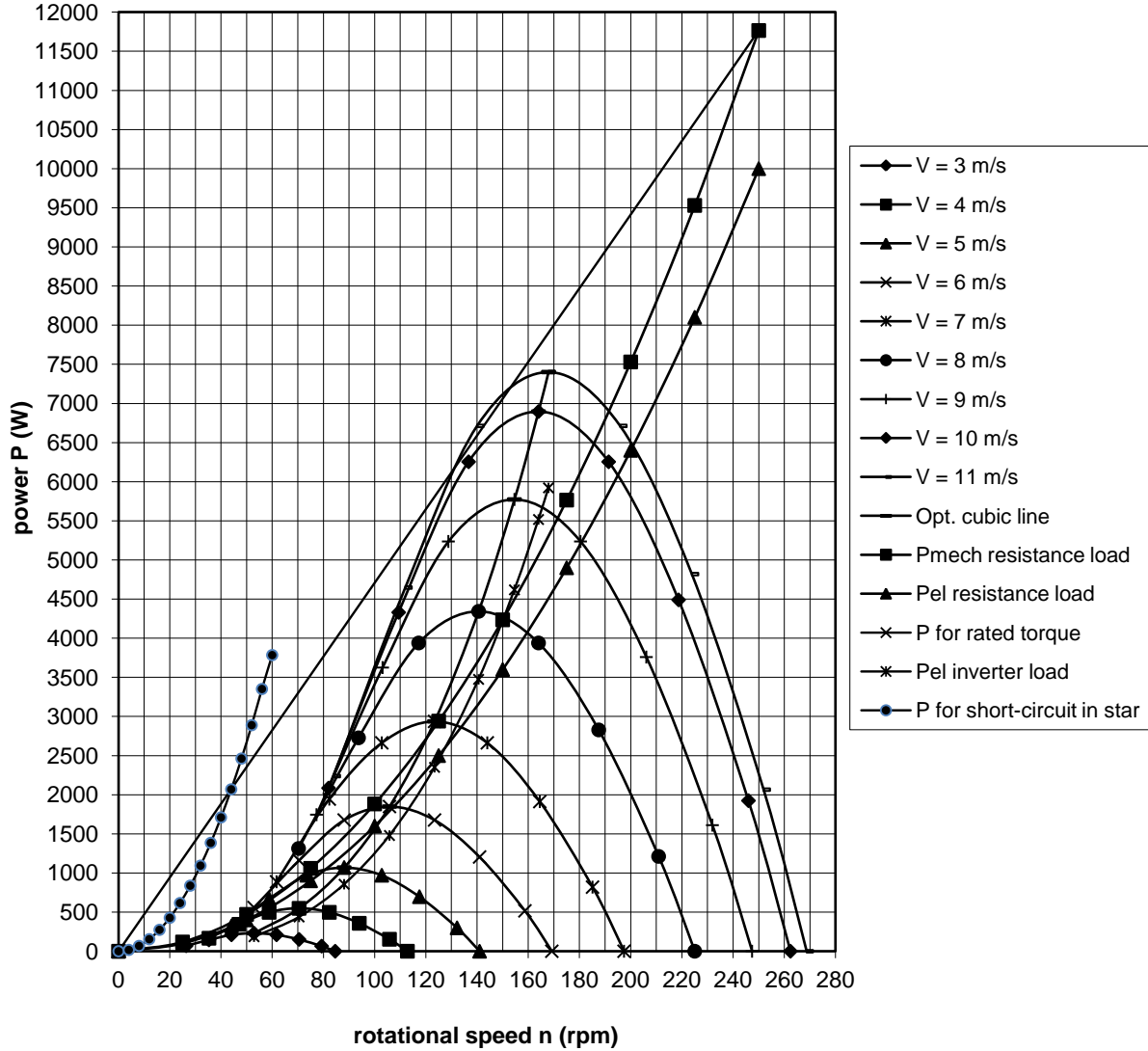


fig. 8 P-n curves of the rotor for different wind speeds and optimum cubic line,  $P_{\text{mech-n}}$  curve and  $P_{\text{el-n}}$  curve of the generator for a resistance load,  $P_{\text{el}}$  for an inverter load

In figure 8 it can be seen that the optimum cubic line is intersecting with the  $P_{\text{mech-n}}$  line for a resistance load at a wind speed of about 7 m/s. So below this wind speed, the load is lower than for the given resistance load if the optimum cubic line is followed. This means that the generator efficiency is somewhat higher than for a resistance load. For wind speeds higher than 7 m/s, the optimum cubic line is lying higher than for the given resistance load and so the efficiency is somewhat lower than for a resistance load. It is assumed that the inverter is programmed such that the optimum cubic line is followed.

So these figures are based on the manufactures specification for a resistance load at  $n = 250$  rpm. The load resistance  $R$  can be calculated if it is assumed that three identical resistors are connected in star to the three phase wires. The voltage over one resistor is equal to the phase voltage  $U_f = 259.8$  V. The line current  $I = 12.83$  A at  $n = 250$  rpm. So according to the law of Ohm, the resistance  $R$  is given by  $R = U / I$  or  $R = 259.8 / 12.83 = 20.25 \Omega$ .

If three resistors are used as load, the winding of one phase is used for all the time to generate power. This power varies according to a  $\sin^2\alpha$  function. The power fluctuation is given in figure 2 of report KD 340 (ref. 9). If a 3-phase winding is rectified in star, only two of the three phases are generating power at the same time. This means that in one phase, power is only generated for  $30^\circ < \alpha < 150^\circ$  and for  $210^\circ < \alpha < 330^\circ$ . This means that no power is generated for  $0^\circ < \alpha < 30^\circ$ , for  $150^\circ < \alpha < 210^\circ$  and for  $330^\circ < \alpha < 360^\circ$ . The loss of generated power because of this effect is about 7 % of the power generated for a resistance load. But this effect is neglected and so it is assumed that the generator is able to generate a DC power of 10 kW at  $n = 250$  rpm.

In the last column of table 3 it can be seen that the heat losses are maximal for  $n = 250$  rpm. This is because the voltage and so also the current decrease at decreasing rotational speed. In figure 7 it can be seen that the torque for a resistance load decreases linear to the decrease of the rotational speed. A PM-generator can also have high torques at low rotational speeds so the chosen value of the resistance gives only a large torque for the rated rotational speed  $n = 250$  rpm. To know the real capacity of the generator, it should also be measured for lower values of the resistance at lower rotational speeds than 250 rpm up to at least the rated torque  $Q = 449$  Nm which is valid for  $n = 250$  rpm. As the current  $I$  is proportional to the torque  $Q$ , the copper losses and so  $P_{\text{heat}}$ , will then be the same for lower rotational speeds and the rated torque. The efficiency will be lower than for the given resistance load but this is acceptable.

A constant rated torque means that the power increases linear to the rotational speed. So a linear P-n curve “rated torque” through  $P = 0$  W and  $n = 0$  rpm and  $P = 11765$  W and  $n = 250$  rpm is also drawn in figure 8. Use of the generator below this line is certainly acceptable without getting a too high value of  $P_{\text{heat}}$ . In figure 8 it can be seen that the optimum cubic line is lying lower than the line for the rated torque for any wind speed. This is an indication that the generator is strong enough for a rotor with a diameter of 6.5 m. So it is assumed that the hinged side vane safety system with a rated wind speed of 11 m/s, is a good choice and that the maximum power is limited enough for this rated wind speed.

The generator efficiency is assumed to be 0.85 for a resistance load. If the optimum cubic line is followed, the real load is lower than the given resistance load if  $V$  is lower than 7 m/s. This means that the efficiency will be somewhat higher than 0.85. The real electrical power depends also on the losses in the rectifier and on the efficiency of the inverter. Rectifier losses are low for high voltages. Modern inverters have a very high efficiency. It is assumed that the total efficiency of generator, rectifier and inverter  $\eta_{\text{tot}} = 0.8$ . The  $P_{\text{el}}$ -n curve for an inverter load such that the optimum cubic line is followed and for a constant efficiency of 0.8 is also given in figure 8. The  $P_{\text{el}}$ -n curve is determined for wind speeds up to 11 m/s. The working point for a certain wind speed is the point of intersection of the P-n curve of the rotor for that wind speed with the optimum cubic line. The electrical power for a certain wind speed is found by going down vertically from the working point until the  $P_{\text{el}}$ -n curve is crossed. The values of  $P_{\text{el}}$  have been determined this way for wind speeds up to 11 m/s and are given in the  $P_{\text{el}}$ -V curve of figure 9. It is assumed that  $P_{\text{el}}$  for higher wind speeds than 11 m/s is the same as the value for  $V = 11$  m/s.

The electrical power is almost 6 kW at a wind speeds of 11 m/s and higher which is very good for a windmill with a rotor diameter of 6.5 m and a rated wind speed of 11 m/s.

If the generated energy is used to power a heat pump, about four times more heat is generated than the input electrical power. So even at moderate wind speeds, a substantial amount of heat will be generated by the VIRYA-6.5.

It is expected that the inverter needs a minimum input voltage to function. So the rotor must have a certain minimal rotational speed. This speed isn't known but at the moment it is supposed that the voltage is too low for wind speeds below 3 m/s. This means that the little energy available in wind speeds below 3 m/s can't be captured. So this is the reason why the  $P_{\text{el}}$ -V curve starts suddenly with  $P_{\text{el}} = 185$  W at  $V = 3$  m/s.

The critical voltage may lie lower and if this is the case, the  $P_{el}$ - $V$  curve starts at a lower wind speed. The  $P_{el}$ - $V$  curve is valid for constant wind speeds and not for average wind speeds. The output for a certain average wind speed is larger than for a certain constant wind speed. This can be demonstrated as follows. Assume we have a constant wind speed of 5 m/s. In the  $P_{el}$ - $V$  curve it can be read that  $P_{el} = 856$  W. Assume we have a wind speed of 7 m/s for one hour and of 3 m/s for one hour. So the average wind speed is 5 m/s. The power for  $V = 3$  m/s is 185 W. The power for  $V = 7$  m/s is 2350 W. The average power is  $(185 + 2350) / 2 = 1268$  W. This is 412 W more or a factor  $1268 / 856 = 1.48$  higher than for a constant wind speed of 5 m/s.

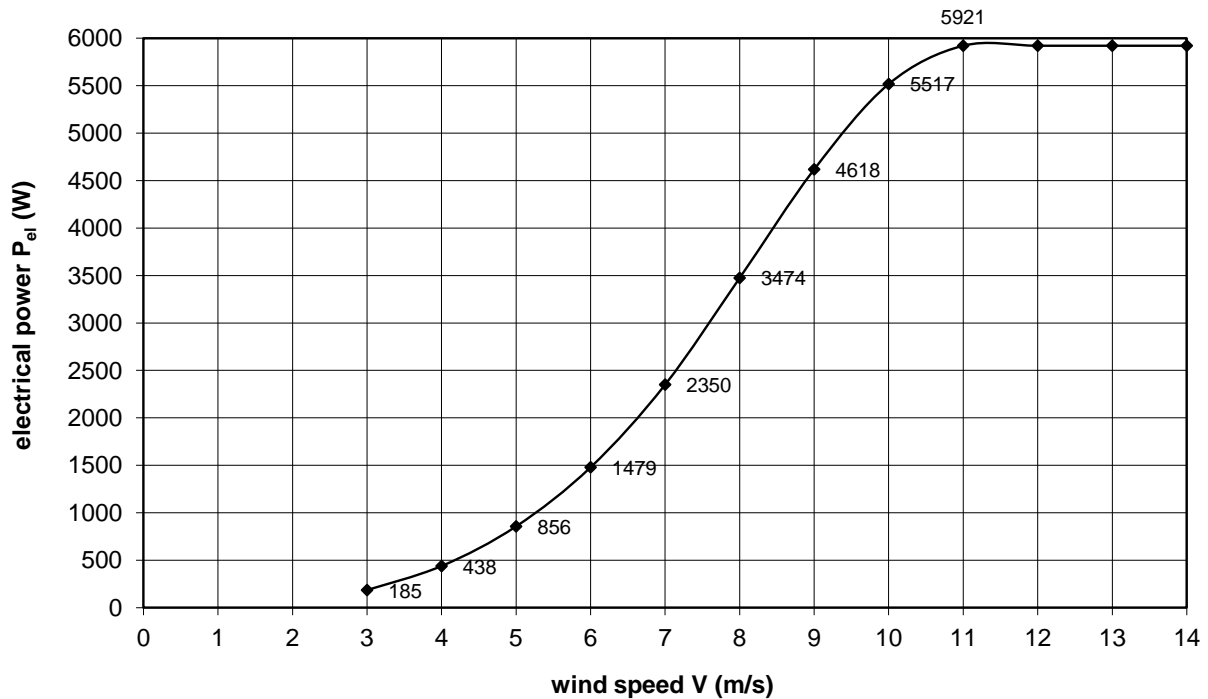


fig. 9  $P_{el}$ - $V$  curve for an inverter load such that the optimum cubic line is followed for  $3 \text{ m/s} < V < 11 \text{ m/s}$

The  $P_{mech}$ - $n$ , the  $P_{el}$ - $n$  curves as given in figure 8 and the  $P_{el}$ - $V$  as given in figure 9 are estimated and not measured. Measured characteristics are more accurate than estimated characteristics. So to be sure that an acceptable matching is realised for the chosen generator of Hefei Top Grand, it is necessary to buy one and to test it at a very large test rig with which it is possible to also measure the torque  $Q$ . One should also select and buy an inverter and measure the real electrical output for grid connection. Finally a complete windmill has to be built and tested with the correct load.

It might be possible to use the rotor as a brake to stop the rotor. To verify if this is possible at any wind speed, one should know the  $P$ - $n$  curve for short-circuit in star but this curve isn't given. The PM-generator which is used for the measurements as given in KD 78 (ref. 10) was measured for different constant voltages rectified in star. The  $Q$ - $n$  curves for 26 V star, 52 V star and 76 V star are given in figure 8 of chapter 4 of KD 78. The  $Q$ - $n$  curve for short-circuit in star before the rectifier is given in figure 4 of chapter 3 of KD 78. If these curves are compared, it can be seen that all curves have about the same shape but that the curve is shifted to the right if the voltage is higher. All curves have about the same maximum value of about 29 Nm. The first part of each curve, up to about 2/3 of the peak value, so up to a torque of about 20 Nm, is about a straight line but the curves bend to the right for higher torques. The curves start at the rotational speed for which the open generator voltage is equal to the average charging voltage. This phenomenon is used to derive the wanted  $P$ - $n$  curve for short-circuit in star for the VIRYA-6.5 generator.



In figure 6 it can be seen that the loaded DC voltage  $U_{DC} = 608 \text{ V DC}$  at  $n = 250 \text{ rpm}$ . The unloaded open DC voltage  $U_{open} = 738 \text{ V}$  at  $n = 250 \text{ rpm}$ . As the U-n curves are straight lines through the origin, it can be read that  $U_{open} = 608 \text{ V DC}$  at  $n = 206 \text{ rpm}$ . This is 44 rpm lower than  $n_{rated} = 250 \text{ rpm}$ . Next it is assumed that Q-n line for a constant voltage of 608 V is a straight line in between the point  $Q = 0 \text{ Nm}$  and  $n = 206 \text{ rpm}$  and the point  $Q = 449 \text{ Nm}$  and  $n = 250 \text{ rpm}$ . This curve is also given in figure 7.

Short-circuit means a constant voltage  $U = 0 \text{ V}$ . So the Q-n curve for short-circuit in star is found by moving the Q-n curve for 608 V DC that much to the left that it intersects with the origin. This means that it must go through the point  $Q = 0 \text{ Nm}$  and  $n = 0 \text{ rpm}$  and the point  $Q = 608 \text{ Nm}$  and  $n = 44 \text{ rpm}$ . This curve is also given in figure 7.

Formula 29 can be written as:

$$P_{mech} = Q * \pi * n / 30 \quad (\text{W}) \quad (30)$$

To determine the P-n curve for short circuit in star, several points have to be chosen on the Q-n curve for short-circuit in star. This was done for every 4 rpm. P is then calculated for every point using formula 30. The result of this procedure is given in table 4. It is assumed that the Q-n curve is about straight for rotational speeds up to 60 rpm and the Q-n and P-n curves are extended up to this rotational speed. For higher rotational speeds than about 60 rpm, the Q-n curve for short-circuit in star will bend to the right and will have a maximum value at a certain rotational speed. This part of the curve can only be determined by measuring. So the P-n curve for short-circuit in star can't be determined for rotational speeds higher than about 60 rpm.

n (rpm)	Q (Nm)	P (W)
0	0	0
4	40.8	17
8	81.6	68
12	122.5	154
16	163.3	274
20	204.1	427
24	244.9	616
28	285.7	838
32	326.5	1094
36	367.4	1385
40	408.2	1710
44	449	2069
48	489.8	2462
52	530.6	2889
56	571.5	3351
60	612.3	3784

table 4 Calculated values of Q and P as a function of n for short-circuit in star

The P-n curve for short-circuit in star can now be derived from table 4 and is also given in figure 8. It can be seen that there is a large distance in between the P-n curve for short-circuit in star and the P-n curve of the rotor for  $V = 11 \text{ m/s}$ . The P-n curve for short-circuit in star couldn't be determined for higher rotational speeds than 60 rpm but by interpolation it can be concluded that the generator can very well be used as a brake to stop the rotor at any wind speed.

Building of a prototype of the VIRYA-6.5 with the chosen PM-generator of Hefei Top Grand is only possible if a composite drawing is made and if detailed drawings are available but I won't make them. So only companies with enough engineering and manufacturing capacity should start with the VIRYA-6.5. The VIRYA-6.5 is certainly not a windmill which can be built by an amateur.

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