Development of the VIRYA-2.8B4, a windmill with a Polycord transmission in between the rotor shaft and the vertical shaft to drive a rope pump or an Archimedian screw pump

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1 Introduction

Up to now Kragten Design has developed thirteen electricity generating windmills with rotor diameters in between 1.2 and 4.6 metre. Eight of these windmills are designed especially for manufacture in developing countries. Although the main focus of Kragten Design is on the development of small electricity generating windmills, there is also a lot of know-how available about water pumping windmills because the designer, Adriaan Kragten, has worked from 1975 until 1990 as a designer of mechanically water pumping windmills in the Wind Energy Group of the University of Technology Eindhoven. In that time the main focus was on the development of windmills driving a single acting piston pump by means of a crank mechanism which was mounted to the rotor shaft. Because the windmill was equipped with a rotor with a rather high tip speed ratio of about 2, this resulted in large dynamic problems. So I have decided to spend no more time to this principle for my engineering office Kragten Design. Kragten Design was founded in 1989 and many KD-reports have been written about aspects of water pumping with windmills using other principles than piston pumps.

Windmills can be used in different ways to pump water and the best choice of windmill and pump depends on many criteria like: the kind of well, the water depth, the required flow, the wind regime, the availability of materials, the workshop facilities, the level of education of manufacturer and user, the training for manufacture and installation, the required maintenance rate and the availability of capital. So there is no one choice which is the best for all situations. However, a windmill for developing countries must be simple and cheap. The maintenance rate must be very low or maintenance must be so simple that it can be done by the owner, otherwise supply of water will stop after failing of the first component.

The idea is to develop the VIRYA-2.8B4 water pumping windmill which has a vertical shaft in the tower which can be coupled to a rope pump or for low heads to an Archimedian screw pump. Both pumps have about the same characteristics because they loose water if they are not working for some time. This means that the starting torque is low from stand still position and therefore a windmill rotor can be used which has a not very high starting torque coefficient. It is expected that the starting torque coefficient of the VIRYA-2.8B4 rotor with λ_d = 2.5 is large enough. As an Archimedian screw pump is rather difficult to manufacture, coupling of the VIRYA-2.8B4 to a rope pump has first priority.

The VIRYA-2.8B4 windmill will be equipped with the so called “hinged side vane safety system” which is used in all VIRYA windmills. This safety system turns the rotor gradually out of the wind at high wind speeds but requires a rather large eccentricity in between the rotor shaft and the tower axis. Therefore a special transmission has to be developed to bridge the distance in between the rotor shaft and the vertical shaft. This transmission must have an accelerating gear ratio to reduce the reaction torque on the head exerted by the vertical shaft. Different transmissions have been examined but the use of a round Polycord string seems to be the most promising solution.

The pump can be coupled to the vertical shaft of the windmill by the same kind of Polycord transmission. This transmission can have a decelerating gear ratio and it can also change the shaft direction from vertical to horizontal for a rope pump or from vertical to an angle of about 30° with the horizon for an Archimedian screw pump. The determination of this second Polycord transmission is out of the scope of this report.

The VIRYA-2.8B4 windmill has a 4-bladed rotor which is provided with 7.14 % curved sheet blades. Two opposite blades are connected to each other by means of a long thin strip and that’s why the rotor is rather flexible. This eliminates vibrations due to wind turbulence and yawing. The advantage of a 4-bladed rotor is that balancing is easy and that a set of two blades and one strip can be transported even mounted. The rotor geometry and strength is calculated in report KD 319 (ref. 1).

2 Characteristics of the windmill rotor

The determination of the Q-n curves of the rotor for different wind speeds is given in figure 4 of chapter 5 of report KD 319 (ref. 1). The Q-n curves are given for a 6 mm plywood vane blade which is so light that the rated wind speed is 8 m/s. Figure 4 of report KD 319 is copied as figure 1. In this figure, n is the rotational speed of the rotor shaft and Q is the rotor torque.

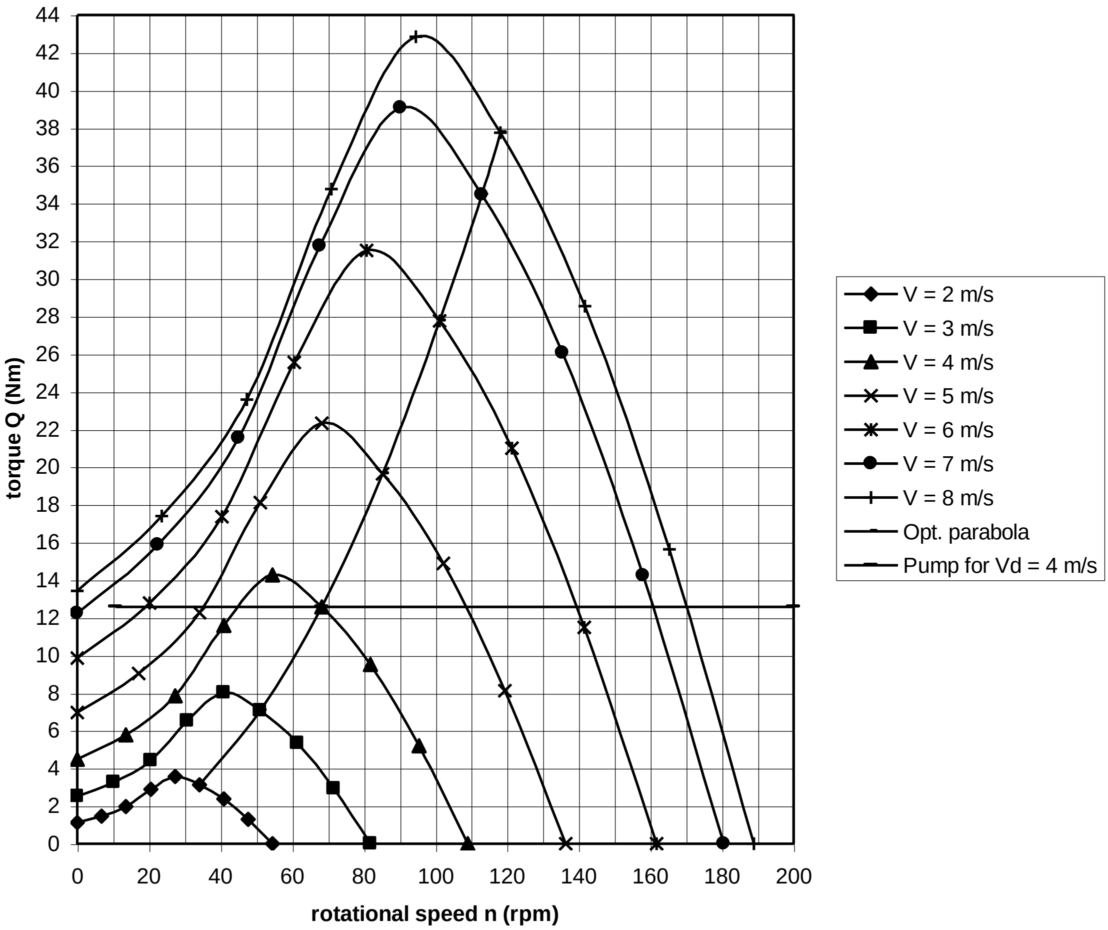

fig. 1 Q-n curves for VIRYA-2.8B4 rotor for a 6 mm plywood vane blade

In figure 1 the optimum parabola is also given. For this curve, the rotor is running at the design tip speed ratio λ_d = 2.5 and it has its maximum power coefficient $C_{p\ max}$ = 0.38.

A rope pump running at moderate speeds, has about a constant pulling force in the rope and therefore needs a constant torque. However, a rope pump has some internal leakage of water along the pistons and this leakage is the cause of the volumetric efficiency. The volumetric efficiency is rather high at high rotational speeds but at a certain very low rotational speed the volumetric efficiency becomes zero which means that no water is pumped. The volumetric efficiency as a function V_p / V_{pd} has been determined in report KD 321 (ref. 2) and is given in figure 1. This figure is copied as figure 2. V_p is the speed of the piston. V_{pd} is the speed of the piston at the design wind speed. The design wind speed is the wind speed at which the working point is lying on the optimum parabola. The working point is the point of intersection of the torque curve of the pump with the Q-n curve of the rotor for a certain wind speed. For figure 2 it was assumed that the volumetric efficiency is 0.842 for the design wind speed.

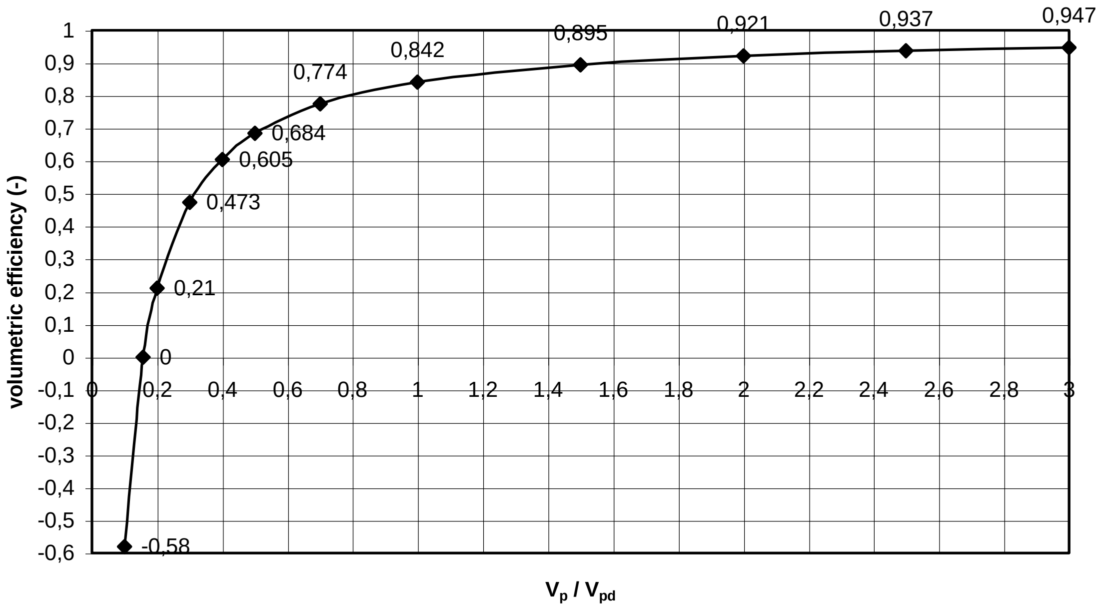

fig. 2 η_{vol} as a function of V_p / V_{pd}

In figure 2 it can be seen that the efficiency curve is zero for about V_p / V_{pd} = 0.158. The piston speed is proportional to the rotational speed n of the rotor so the efficiency will also be zero for n / n_d = 0.158.

Now suppose that the gear ratios and the pump dimensions and the water height are chosen such that the design wind speed is 4 m/s. This means that the torque curve of the pump seen on the rotor shaft is a horizontal line which goes through the point of intersection of the optimum parabola and the P-n curve of the rotor for V = 4 m/s. This point is lying at a rotational speed of 68.2 rpm (see figure 1). The rotational speed where the volumetric efficiency becomes zero will lie at a rotational speed of 0.158 * 68.2 = 10.8 rpm. So above this rotational speed, water is pumped and the torque is constant. The horizontal torque line of the pump seen on the rotor shaft for V_d = 4 m/s is also drawn in figure 1. It can be seen that the beginning point of the line at n = 10.8 rpm is lying at about the estimated the P-n curve of the rotor for V = 6.5 m/s.

The first impression therefore is that a wind speed of 6.5 m/s is required to start the rotor. However, this is not true because, if the rotor is standing still for a certain time, all the water in the rising main is leaked back into the well. So for this condition the pump torque is zero. The pump torque increases proportional to the water level in the rising main. Now suppose that the wind speed increases suddenly from 0 to 4 m/s. In figure 1 it can be seen that the starting torque of the rotor for V = 4 m/s is 4.47 Nm and that the torque is rising directly if the rotational speed increases. So the rotor will start and almost the whole torque will be used for acceleration of the rotor. But during this acceleration, the water level in the rising main will rise and the required pump torque will rise too.

In figure 1 it can be seen that the Q-n line of the pump has a left point of intersection with the Q-n line of the rotor for V = 4 m/s at a rotational speed of 45 rpm. In report KD 321 it is found that the rotor accelerates so fast that it reaches this rotational speed before the water in the rising main has reached the maximum level. The real starting wind speed will therefore be much lower than 6.5 m/s and it is expected that the rotor will start at least at a wind speed equal to the design wind speed which was chosen 4 m/s. This is acceptable.

It is possible to transform the Q-n curves of the rotor to the vertical shaft or to the pump shaft if the gear ratios and the transmission efficiencies are known. This will be done in chapter 6.

3 Determination of the transmission in between rotor shaft and vertical shaft

All VIRYA windmills have an eccentricity which is in between about 8 % and 10 % of the rotor diameter. This large eccentricity is required to realise that the rotor thrust is having the largest influence on the rotor moment M_{rot} which turns the rotor out of the wind and that other moments caused by the side force on the rotor, the self orientating moment, and the head bearing friction have only a relatively small influence.

If the windmill has a rotating vertical shaft, the reaction torque M_{react} in this shaft, which works in a direction opposite to the direction of rotation, has also an influence on the safety system. As this influence must be relatively small, the rotor eccentricity must be at least 10 % of the rotor diameter to realise a smooth functioning of the safety system. This means that the eccentricity e must be at least 0.28 m for 2.8 m rotor diameter.

The head geometry and the transmission can be chosen such that M_{react} works in the same or in the opposite direction as M_{rot}. If both moments work in the same direction, the total moment will be rather high and this requires a rather large vane blade area to give the balancing moment. Therefore, the head geometry and the transmission will be chosen such that both moments work in opposite direction.

The choice of counteracting moments has an extra advantage that the resulting moment becomes larger if the pump load becomes zero, for instance because the well is empty. This means that an unloaded rotor will turn more out of the wind than a loaded rotor and this favours safety.

There is a reason to favour right hand rotation of the vertical shaft seen from the top of the windmill. If the vertical shaft is made from different pieces, these pieces can be coupled by simple thread couplings without a tendency to unscrew. This means that the reaction moment on the head is left hand seen from the top of the windmill. The fact that the rotor moment has to work opposite the reaction moment, means that the rotor moment must be right hand. This requires a rotor which is mounted to the left side of the tower axis if one is looking to the front of the rotor. This positioning is opposite the positioning of all other VIRYA windmills but this is no problem for a new windmill.

The VIRYA-2.8B4 has a rotor which is mounted to the shaft by means of a tapered shaft end and one central bolt. No key is used. To prevent coming loose of this connection, it is required that the rotor is turning right hand if a normal bolt with right hand thread is used.

These requirements result in a special path for the round string and a special positioning of the string wheels. It appears to be possible to design a transmission with only three wheels, one large wheel on the rotor shaft, one small wheel on the vertical shaft and one small auxiliary wheel.

Next the transmission itself has to be determined. The basic transmission element is a round Polycord string of the Swiss manufacture Habasit. A data sheet (in German) has been added as appendix 1. The maximum available string diameter is 15 mm but a 12 mm string is chosen because a 15 mm string would result in very large wheel diameters. The Polycord string is supplied endless and is melted to the correct length using a quick-melt mirror. This can easily be done if a quick-melt mirror is available or one can order an already welded string at the supplier. The string has a pre-tension of about 8 % by taking the string length 8 % shorter than the theoretical value, so the wheels can have fixed position. The wheels have 60° V-grooves which are rounded at the bottom with about the radius of the string and at the outside to prevent damage of the string at mounting. The correct path means that the direction of the belt as it enters or leaves a wheel is exactly tangential so there is no tendency for the belt to run out off the wheel groove.

Another requirement is that the wheels don't touch each other. Several options for the positioning of the auxiliary wheel exist but I favour the position where the spanned bow on the big wheel is 270° and where the spanned bow on the small wheel is 225°. For these angles, the spanned bow on the auxiliary wheel is 90°. The shaft of the auxiliary wheel is horizontal and makes an angle of 45° with the rotor shaft.

The angle in between the vane arm and the rotor shaft is also 45° and therefore the shaft of the auxiliary wheel is parallel to the vane arm. This facilitates the connection of the auxiliary shaft to the head frame. This positioning of the auxiliary wheel is possible for accelerating gear ratios of 2.5 and larger. If the gear ratio is chosen smaller than 2.5, the auxiliary wheel will touch the large wheel on the rotor shaft. A gear ratio of 2.5 is chosen because larger gear ratios result in too high a rotational speed of the vertical shaft. A front and top view of the string path and the three wheels is given in figure 3.

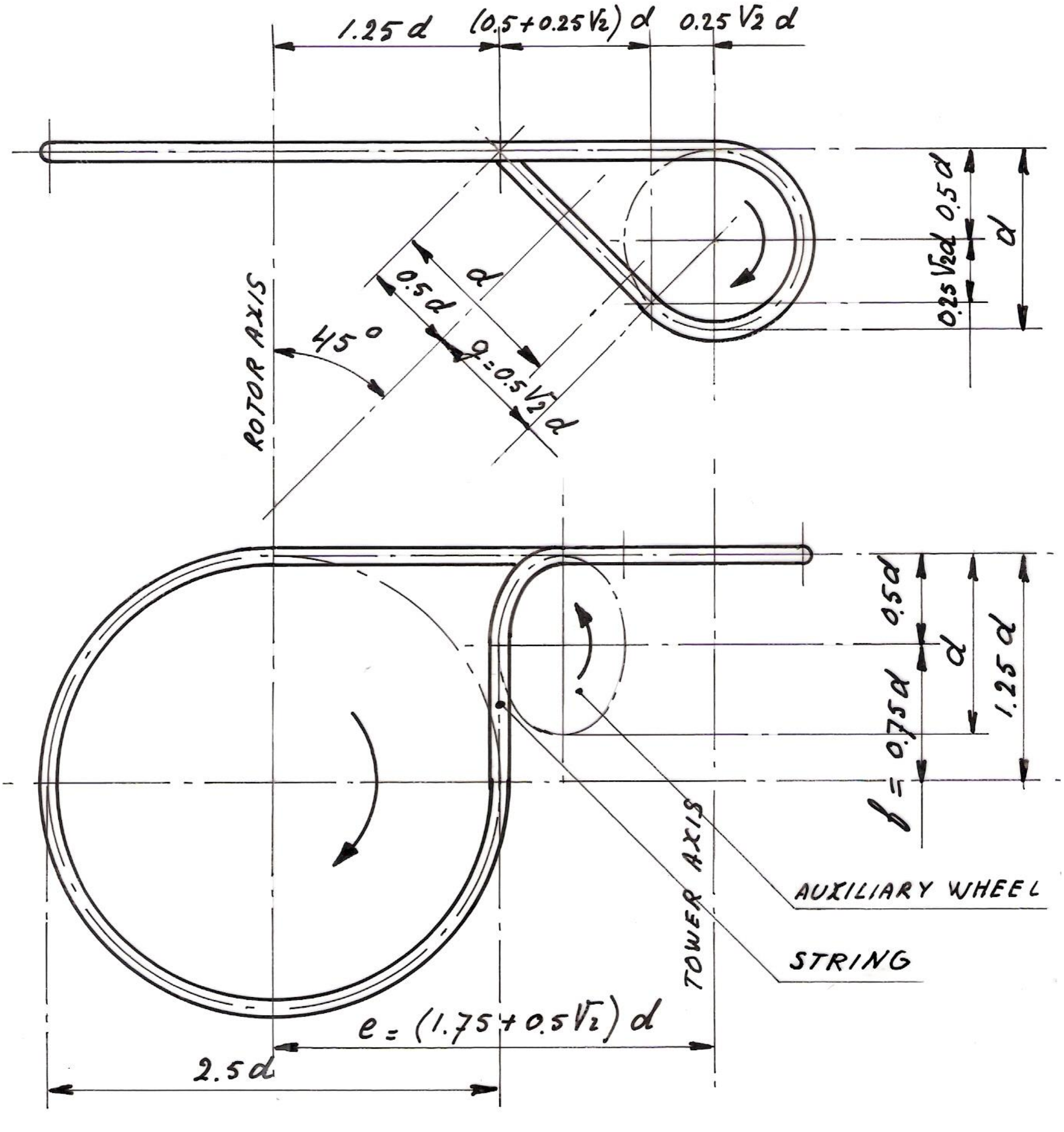

fig. 3 String path and wheels for VIRYA-2.8B4 transmission with i = 2.5

The pitch circle is lying at the hart of the string. As the string has a diameter of 12 mm, the inside diameter of the groove in the wheels is 12 mm smaller than the pitch diameter. The outside diameter is 2 * 9 = 18 mm larger than the inside diameter, so 18 – 12 = 6 mm larger than the pitch diameter. In the Polycord folder (see appendix 1) it is specified that the minimum wheel diameter for a 12 mm string is 120 mm. Assume we take d = 120 mm. The pitch diameter D of the big wheel on the rotor shaft is given by:

$$D = i * d \qquad \text{(mm)} \qquad (1)$$

Substitution of i = 2.5 and d = 120 mm in formula 1 gives D = 300 mm. It can be calculated for the eccentricity e for the chosen positioning of the wheels and i = 2.5 that:

$$e = (1.75 + \tfrac{1}{2}\sqrt{2}) * d \qquad \text{(m)} \qquad (2)$$

Substitution of d = 120 mm in formula 1 gives e = 295 mm = 0.295 m. This is 10.5 % of the rotor diameter so the requirement that the eccentricity is at least 10 % of the rotor diameter is fulfilled.

If the rotor shaft runs clock wise, the loaded part of the string is the part which passes the auxiliary wheel but this seems to be no problem. The unloaded part is the part which goes straight from the big wheel on the rotor shaft to the small wheel on the vertical shaft and therefore there is no tendency to run out of the groove in the small wheel even if the pre-tension is almost gone. Next the resulting moment M_{res} of the rotor and the vertical shaft will be calculated. If the rotor is perpendicular to the wind, only the rotor thrust F_t contributes to M_{rot}. For this situation M_{rot} is given by:

$$M_{rot} = e * C_t * \tfrac{1}{2}\rho V^2 * \pi R^2 \qquad \text{(Nm)} \qquad (3)$$

C_t is the thrust coefficient which is about 0.75 for the chosen rotor. The reaction moment M_{react} depends on the rotor torque Q, the gear ratio i and the transmission efficiency η_{tr} (not taken in % but as a factor of 1) and is given by the formula:

$$M_{react} = Q * \eta_{tr} / i \qquad \text{(Nm)} \qquad (4)$$

The rotor torque Q for a rotor perpendicular to the wind is given by formula 4.3 of KD 35 (ref. 3). This formula is copied as formula 5.

$$Q = C_q * \tfrac{1}{2}\rho V^2 * \pi R^3 \qquad \text{(Nm)} \qquad (5)$$

(4) + (5) gives:

$$M_{react} = \eta_{tr} * C_q * \tfrac{1}{2}\rho V^2 * \pi R^3 / i \qquad \text{(Nm)} \qquad (6)$$

For the VIRYA-2.8B4, both moments work in different direction, so:

$$M_{res} = M_{rot} - M_{react} \qquad \text{(Nm)} \qquad (7)$$

(3) + (6) + (7) gives:

$$M_{res} = \tfrac{1}{2}\rho V^2 * \pi R^2 (e * C_t - \eta_{tr} * C_q * R / i) \qquad \text{(Nm)} \qquad (8)$$

This formula is required for the determination of the vane blade and vane arm dimensions. These calculations will be given in chapter 6.

(3) + (6) gives:

$$M_{rot} / M_{react} = e * C_t * i / (\eta_{tr} * C_q * R) \qquad (-) \qquad (9)$$

The transmission efficiency η_{tr} is supposed to be 0.95. It is supposed that the rotor runs at the design tip speed ratio λ_d = 2.5 and at its optimum C_q value $C_{q\ opt}$ = 0.152 (see fig. 2, KD 319). Substitution of e = 0.2949 m, C_t = 0.75, i = 2.5, η_{tr} = 0.95, C_q = $C_{q\ opt}$ = 0.152 and R = 1.4 m in formula 9 gives M_{rot} / M_{react} = 2.735. So M_{rot} gives the main contribution to M_{res}. So the requirement that the reaction moment is only a small part of the total moment is also fulfilled.

4 Determination of the string dimensions for the upper transmission

The maximum moment which can be passed through for a certain string diameter and a certain wheel diameter depends on the angle of the spanned bow. The smallest angle for a loaded wheel is 225° for the small wheel on the vertical shaft.

The allowable net pulling force F_{net} is given in the first table (Tafel 1) of annex 1 (in German) as Nennumfangskraft F_{un}). F_{net} for a 12 mm diameter string is 145 N. This value is valid for a spanned bow (Umschlingungswinkel) β = 180°. In figure 1 of appendix 1, the bow factor (Winkelfaktor) c_1 is given as a function of the spanned bow β. For a spanned bow of 225° we can read that c_1 is 0.87 (by interpolation just below β = 220°). The given value of F_{net} for 180° has to be divided by c_1 to find the corrected value $F_{net\ cor}$ or in formula:

$$F_{net\ cor} = F_{net} / c_1 \qquad (N) \qquad (10)$$

Substitution of F_{net} = 145 N and c_1 = 0.87 in formula 10 gives $F_{net\ cor}$ = 167 N.

The force for 8 % pre-tension = 200 N (see upper table appendix 1). This means that the total pulling force for the loaded part of the string is 200 + ½ * 167 = 283.5 N and that the pulling force for the unloaded part of the string is 200 – ½ * 167 = 116.5 N. So the unloaded part has still a considerable large pulling force and will therefore not run out of the wheels. This is an indication that even for a higher torque than the allowable torque, there is a large reserve. May be the string will slip over the smallest wheel if the torque is too large. The string will never break because the breaking force is 4500 N.

The maximum rotor torque Q_{max} for normal maximum pulling force is given by:

$$Q_{max} = F_{netcor} * D / 2 \qquad (Nm) \qquad (11)$$

Substitution of $F_{net\ cor}$ = 167 N and D = 300 mm = 0.3 m in formula 18 gives Q_{max} = 25.1 Nm. In figure 1 it can been seen for V = 8 m/s that the maximum torque for the optimum parabola is 37.74 Nm and that the absolute maximum torque is 42.83 Nm for λ = 2. So the string transmission seems to be not strong enough to brake the rotor to stand still using a disk brake on the vertical shaft. In figure 1 it can be seen that the starting torque for V = 8 m/s (and higher) is 13.41 Nm. So it seems possible to first turn the head out of the wind (by a rope connected to the vane arm) and then lock the vertical shaft. Another option for stopping of the rotor might be development of a construction with which the vane blade can be put in the horizontal position.

In figure 1 the pump torque is given for a design wind speed of 4 m/s. This line is lying at a torque level of 12.58 Nm. This torque is much smaller than Q_{max} = 25.1 Nm for the given string and wheel diameters. A design wind speed of 5 m/s results in a torque level of 19.65 Nm and even this value is smaller than 25.1 Nm. A higher design wind speed will surely not be chosen so the geometry of the string and the wheels seems to be acceptable.

The string length l is determined by first calculating the theoretical string l_{th} length and than compensate l_{th} for the required pre-tension. The theoretical string length is the sum of three curved parts a, b and c and three straight parts e, f and g. d is the pitch diameter of the two small wheels. a is the spanned bow over the big wheel. b is the spanned bow over the small wheel on the vertical shaft. c is the spanned bow over the auxiliary wheel. e is the horizontal string length in between the small wheel and the big wheel (which is the same as the eccentricity e). f is the vertical string length in between the big wheel and the auxiliary wheel. g is the horizontal string length in between the auxiliary wheel and the small wheel. The diameter D of the big wheel is 2.5 d. The values of e, f and g are given in figure 2 as a function of d. This gives:

$l_{th} = a + b + c + e + f + g$ (mm) (12)

$a = 270/360 * \pi * 2.5 * d = 5.89049\ d$ (mm) (13)

$b = 225/360 * \pi * d = 1.96350\ d$ (mm) (14)

$c = 90/360 * \pi * d = 0.78540\ d$ (mm) (15)

$e = (1.75 + 0.5\sqrt{2}) * d = 2.45711\ d$ (mm) (16)

$f = 0.75\ d$ (mm) (17)

$g = 0.5\sqrt{2} * d = 0.70711\ d$ (mm) (18)

(12) + (13) + (14) + (15) + (16) + (17) + (18) gives:

$l_{th} = 12.5536\ d$ (mm) (19)

Substitution of d = 120 mm in formula 19 gives l_{th} = 1506.4 mm. If the pre-tension is 8 % it means that $l_{th} = 1.08 * l$. This gives:

$l = l_{th} / 1.08$ (mm) (20)

Substitution of l_{th} = 1506.4 in formula 27 gives l = 1394.8 mm. Because of the melting process some mm of the string will be melted which shortens the string. Take therefore l = 1396 mm. The melted material can be removed with a file after cooling down of the string.

5 Determination of the vertical shaft.

In figure 1 it can be seen that the maximum unloaded rotational speed of the rotor is about 189 rpm for V = 8 m/s or higher. The rotational speed of the vertical shaft n_v is given by:

$n_v = n * i$ (rpm) (21)

In practice the gear ratio may vary a little depending on the load because there will be some slip of the string around the wheels but this effect is neglected. Substitution of $n = n_{max}$ = 189 rpm and i = 2.5 in formula 21 gives $n_{v\ max}$ = 472.5 rpm. This is rather high and the vertical shaft therefore has to be checked for “instability” (the Dutch word is zwiep, the correct English word could not be found). Instability for a rotating shaft can be compared to buckling for a compressed rod. In practice the maximum pump speed will normally be lower than the calculated value because of the pump load. In figure 1 it can be seen that the maximum loaded rotational speed is 170 rpm for V_d = 4 m/s.

However, it is possible that the pump load disappears, for instance if there is no water. Therefore the worst case of an unloaded rotor has to be taken for instability calculations.

Instability of a rotating shaft can be understood best for a long horizontal shaft turning in two bearings. The shaft will bend downwards in the middle because of the weight of the shaft itself. Therefore a certain distance will be created between the centre of gravity and the real rotational axis. Because of this distance a centrifugal force exists which has a tendency to increase the distance. At a certain critical rotational speed n_c, the stiffness of the shaft is no longer large enough to prevent rapid growth of the distance and this results in a sudden break out of the shaft which is very dangerous. If the shaft is not horizontal but vertical, the problem is not initiated by the weight of the shaft but by sudden vibrations or if the shaft is bent somewhat. The shaft is normally calculated for horizontal position even if it is vertical in practice. However, I think that there is more reserve if the shaft is vertical. The instability is influenced by the stiffness of the bearings. It is assumed that the bearings have no stiffness and the given formulas are valid for this situation. The critical rotational speed n_c is given by:

$$n_c = 30/\pi * \sqrt{(g / f)} \qquad \text{(rpm)} \qquad (22)$$

In this formula is g is the acceleration of gravity (m/s^2) and f is the deflection (m) because of the weight of the shaft. For a shaft with length l (m) and an equal spread load q (caused by the weight), the deflection f is given by:

$$f = 5\, q * l^4 / 384\, EI \qquad \text{(m)} \qquad (23)$$

For a shaft with diameter d (m) and a density ρ_s (kg/m^3), the weight per length q is given by:

$$q = \pi/4\, d^2 * \rho_s * g \qquad \text{(N/m)} \qquad (24)$$

The moment of inertia I (m^4) of a round shaft with diameter d (m) is given by:

$$I = \pi/64\, d^4 \qquad (m^4) \qquad (25)$$

(23) + (24) + (25) gives:

$$f = 5/24\, \rho_s * g * l^4 / (E * d^2) \qquad \text{(m)} \qquad (26)$$

(22) + (26) gives:

$$n_c = 20.92 * d/l^2 * \sqrt{(E / \rho_s)} \qquad \text{(rpm)} \qquad (27)$$

Assume we take a steel shaft. This gives $\rho_s = 7.8 * 10^3$ kg/m^3 and $E = 2.1 * 10^{11}$ N/m^2. The 6 m long lowest tower part has horizontals at every 0.5 m so it is easy for connection of the bearings if l is a whole number times this value. The value for d must be taken as small as possible because the vertical shaft must pass through a hole in the head pin. Assume d = 12 mm = 0.012 m and l = 1.5 m. Substitution of these values in formula 27 gives n_c = 579 rpm. This is larger than the calculated maximum rotational speed of 472.5 rpm so the chosen distance of 1.5 m in between the bearings is acceptable for a 12 mm shaft. The upper tower part is made of a 2 m long pipe and it is difficult to make a bearing somewhere in the pipe. However, the shaft will have bearings in the top and in the bottom of the head pin and therefore the upper part of the shaft is fixed. This allows a much longer shaft length.

The advantage of d = 12 mm is that metric thread is available for this diameter so the shaft ends can be connected to each other by long nuts. For a shaft diameter of 12 mm, ball bearings can be used for the shaft bearing .

The torsion stress for a 12 mm shaft will be calculated for the maximum moment which is allowed for the Polycord transmission which is 25.1 Nm. It is assumed that connections in between shaft ends are made by long M12 nuts so the shaft is weakened by thread. Assume that the effective inside diameter of the thread is 10 mm. Substitution of $Q = Q_{max} = 25.1$ Nm, $\eta_{tr} = 0.95$ and $i = 2.5$ in formula 4 gives $Q_{react} = 9.54$ Nm = 9540 Nmm.

The torsion stress τ is calculated by formula:

$$\tau = Q / (\pi/16 * d^3) \quad (N/mm^2) \tag{28}$$

Substitution of $Q = Q_{max} = 9540$ N/mm^2 and d = 10 mm in formula 28 gives $\tau = 48.6$ N/mm^2. This is a low stress which is surely acceptable, even if stainless steel is used as material for the vertical shaft.

6 Determination of the vane dimensions

The hinged side vane safety system is described in report KD 223 (ref. 4) for the VIRYA-3.3D windmill. In this report certain assumptions have been made for the self orientating moment which are based on measurements performed on a rotor with a design tip speed ratio of 6. The VIRYA-2.8 has a design tip speed ratio of 2.5. Recently a scale model of a rotor with $\lambda_d = 2.5$ has been made and it was found that this rotor had no self orientating moment. The side force on the rotor will be very small for small yaw angles so the rotor moment is only determined by the rotor thrust. For calculation of the vane geometry it is assumed that the rotor is perpendicular to the wind. This means that formula 8 can be used for the resulting moment M_{res} of the rotor moment M_{rot} and the reaction moment of the vertical shaft M_{react}.

The rotor is only perpendicular to the wind at low wind speeds. For low wind speeds, the vane blade is hanging almost vertical and this means that the normal force on the vane blade is large if compared to the aerodynamic force on the vane arm. For this condition, the aerodynamic force on the vane arm can be neglected so only the normal force N on the vane blade is taken into account. The vane moment around the tower axis M_{vt} produced by N for low wind speeds is given by formula 30 of KD 223 which is copied as formula 29.

$$M_{vt} = C_n * ½\rho V^2 * h * w * \cos\theta * (R_v + i_1) \quad (Nm) \tag{29}$$

In this formula C_n is the normal coefficient for a square vane (-), ρ is the air density kg/m^3, V is the wind speed at the vane blade (m/s). Because the vane blade juts out beside of the rotor it is the same as the undisturbed wind speed V. h is the height of the vane blade (m). w is the width of the vane blade (m). θ is the angle in between the vane blade and the vertical, R_v is the distance in between the front side of the vane blade and the tower centre measured in a direction parallel to the vane blade axis and i_1 is the distance in between the normal force N and the front side of the vane blade.

C_n depends on the angle α in between the vane blade and the wind direction. The C_n-α curve for a square vane blade is given in figure 6 of KD 223. For a square vane blade, h and w are the same. For low wind speeds θ is small which means that $\cos\theta$ is almost 1. R_v depends on the chosen length of the vane arm pipe. i_1 depends on the blade width w and on the angle of attack α in between the vane blade and the wind direction. This angle depends of the angle ϕ_2 in between the hinge axis and the rotor axis and on the yaw angle δ. $\alpha = \phi_2$ for $\delta = 0°$ (see KD 223 figure 1). ϕ_1 is chosen 45°. ϕ_2 is chosen 30° for all electricity generating VIRYA windmills but it must be chosen smaller for a windmill with a rotating vertical shaft to prevent stalling of the vane blade if the reaction torque of the vertical shaft becomes zero. It is chosen that $\phi_2 = 22°$, so $\alpha = 22°$ for $\delta = 0°$. $i_1 = 0.35$ w for $\alpha = 22°$ (see KD 223, figure 7).

To realise a low rated wind speed of 8 m/s, a light 2 mm aluminium vane blade is used. This material is available in sheet of 1.5 * 3 m. With no material waste it is possible to make eight vane blades of 0.75 * 0.75 m from one standard sheet. Assume this vane blade size is chosen, so w = 0.75 m. This results in i_1 = 0.35 * 0.75 = 0.263 m. If the rotor is perpendicular to the wind at low wind speed M_{res} must be the same as M_{vt}, so:

$$M_{res} = M_{vt} \qquad (30)$$

(8) + (29) + (30) gives:

$$\pi R^2 (e * C_t - \eta_{tr} * C_q * R / i) = C_n * h * w * \cos\theta * (R_v + i_1) \qquad (31)$$

Formula 31 can be written as:

$$C_n = \pi R^2 (e * C_t - \eta_{tr} * C_q * R / i) / h * w * \cos\theta * (R_v + i_1) \qquad (32)$$

The only value which still has to be chosen is R_v. R_v must be taken so long that the vane blade juts out of the rotor plane for a rotor with R = 1.4 m. The vane arm makes an angle ϕ_1 = 45° with the rotor shaft. This means that the length of the vane arm must be at least $\sqrt{2}$ * R - e = $\sqrt{2}$ * 1.4 – 0.295 = 1.685 m. R_v is not measured in the direction of the vane arm but in the direction of the vane blade. The angle in between the vane axis and the vane arm is $\phi_1 - \phi_2$ = 45° – 22° = 23° and the minimum value for R_v has therefore to be 1.685 * cos23° = 1.551 m. Assume R_v = 1.58 m. Substitution of R = 1.4 m, e = 0.295 m, C_t = 0.75, η_{tr} = 0.95, C_q = 0.152, i = 2.5, h = 0.75 m, w = 0.75 m, cosθ = 1, R_v = 1.58 m and i_1 = 0.263 m in formula 32 gives that C_n = 0.834. In figure 6 of KD 223 it can be seen that C_n = 0.834 is realised for about α = 18°. As α = 22° for the rotor perpendicular to the wind, an angle α = 18° will be realised for δ = -4° which is a good choice.

The reaction torque is zero if the well is empty, if the rope of the rope pump is broken or if all the water is flown down in the well after a long stand still of the rotor. Also in this case the rotor must be about perpendicular to the wind at low wind speeds. If the reaction torque is zero it means that C_q = 0. Substitution of R = 1.4 m, e = 0.295 m, C_t = 0.75, η_{tr} = 0.95, C_q = 0, i = 2.5, h = 0.75 m, w = 0.75 m, cosθ = 1, R_v = 1.58 m and i_1 = 0.263 m in formula 32 gives that C_n = 1.314. In figure 6 of KD 223 it can be seen that C_n = 1.314 is realised for about α = 29°. As α = 22° for the rotor perpendicular to the wind, an angle α = 29° will be realised for δ = 7°. So the rotor is still about perpendicular to the wind if the reaction torque of the vertical shaft is zero.

Formula 32 is only valid if the rotor is perpendicular to the wind. The rotor thrust is reduced by a factor $\cos^2\delta = \cos^2 7° = 0.985$ but as this is almost 1, the reduction of the rotor thrust by a yaw angle δ = 7° can be neglected and so formula 32 can still be used.

7 Determination of the Q_v-n_v curves

It is easy to transform the Q-n curves on the rotor shaft as given in figure 1 to the vertical shaft. The torque in the vertical shaft Q_v for low wind speeds is the same as the reaction moment M_{react} given by formula 6 but has the opposite direction. But for higher wind speeds, the safety system causes a certain yaw angle δ. In stead of V one has therefore to use V cosδ. This results in the following formula for $Q_{v\delta}$:

$$Q_{v\delta} = \eta_{tr} * C_q * \tfrac{1}{2}\rho V^2 * \cos^2\delta * \pi R^3 / i \quad \text{(Nm)} \qquad (33)$$

The rotational speed of the vertical shaft is given by formula 21. The rotational speed for a yawing rotor is given by formula 7.1 of KD 35 which is copied as formula 35:

$n_\delta = 30 * \lambda * \cos\delta * V / \pi R$ (rpm) (34)

(21) + (34) gives:

$n_{v\delta} = 30 * \lambda * i * V * \cos\delta / \pi R$ (rpm) (35)

Substitution of $\eta_{tr} = 0.95$, $\rho = 1.2$ kg/m^3, R = 1.4 m and i = 2.5 in formula 33 gives:

$Q_{v\delta} = 1.9655\, C_q * V^2 * \cos^2\delta$ (Nm) (36)

Substitution of i = 2.5 and R = 1.4 m in formula 35 gives:

$n_{v\delta} = 17.0523\, \lambda * V * \cos\delta$ (rpm) (37)

The C_q-λ curve is given in figure 2 of KD 319. The estimated δ-V curve is given in figure 3 of KD 319. The Q_v-n_v curves for i = 2.5 are now determined using formulas 36 and 37 for the same choice of λ and C_q and the same values of δ as has been used to make figure 1. The result of the calculations are given in table 1. The Q_v-n_v curves and the optimum parabola are given in figure 4.

		V = 2 m/s $\delta = 0°$		V = 3 m/s $\delta = 0°$		V = 4 m/s $\delta = 0°$		V = 5 m/s $\delta = 0°$		V = 6 m/s $\delta = 8°$		V = 7 m/s $\delta = 19°$		V = 8 m/s $\delta = 30°$	
λ (-)	C_q (-)	n_v (rpm)	Q_v (Nm)	n_v (rpm)	Q_v (Nm)	n_v (rpm)	Q_v (Nm)	n_v (rpm)	Q_v (Nm)	$n_{v\delta}$ (rpm)	$Q_{v\delta}$ (Nm)	$n_{v\delta}$ (rpm)	$Q_{v\delta}$ (Nm)	$n_{v\delta}$ (rpm)	$Q_{v\delta}$ (Nm)
0	0.054	0	0.42	0	0.96	0	1.70	0	2.65	0	3.78	0	4.65	0	5.09
0.5	0.07	17.1	0.55	25.6	1.24	34.1	2.20	42.6	3.44	50.7	4.90	56.4	6.03	59.1	6.60
1	0.095	34.1	0.75	51.2	1.68	68.2	2.99	85.3	4.67	101.3	6.66	112.9	8.18	118.1	8.96
1.5	0.14	51.2	1.10	76.7	2.48	102.3	4.40	127.9	6.88	152.0	9.81	169.3	12.05	177.2	13.21
2	0.1725	68.2	1.36	102.3	3.05	136.4	5.42	170.5	8.48	202.6	12.09	225.7	14.85	236.3	16.27
2.5	0.152	85.3	1.20	127.9	2.69	170.5	4.78	213.2	7.47	253.3	10.65	282.2	13.09	295.4	14.34
3	0.115	102.3	0.90	153.5	2.03	204.6	3.62	255.8	5.65	304.0	8.06	338.6	9.90	354.4	10.85
3.5	0.0629	119.4	0.49	179.0	1.11	238.7	1.98	298.4	3.09	354.6	4.41	395.0	5.42	413.5	5.93
4	0	136.4	0	204.6	0	272.8	0	341.0	0	405.3	0	451.5	0	472.6	0

table 1 Calculated values of n_v and Q_v as a function of λ and V for the VIRYA-2.8B4 rotor

For the reducing gearing to the shaft of the rope pump, the modified Q_p-n_p curves for the pump shaft can be determined in the same way as it is done for the vertical shaft. Only the gear ratio and the efficiency for this second step has to be known. If the reducing gear ratio is taken 1 / 2.5 = 0.4 one will find about the same Q-n curves as for the rotor shaft but because of the efficiencies of both transmissions the torque levels will be a little lower.

The determination of the reducing gearing to the shaft of the rope pump is without the scope of this report. The vertical shaft needs probably a double gearing at the bottom side to stabilise the position of the transmission wheel. Detailed information about the construction of the head, the transmission and the vertical shaft bearings can only be given after making a composite drawing.

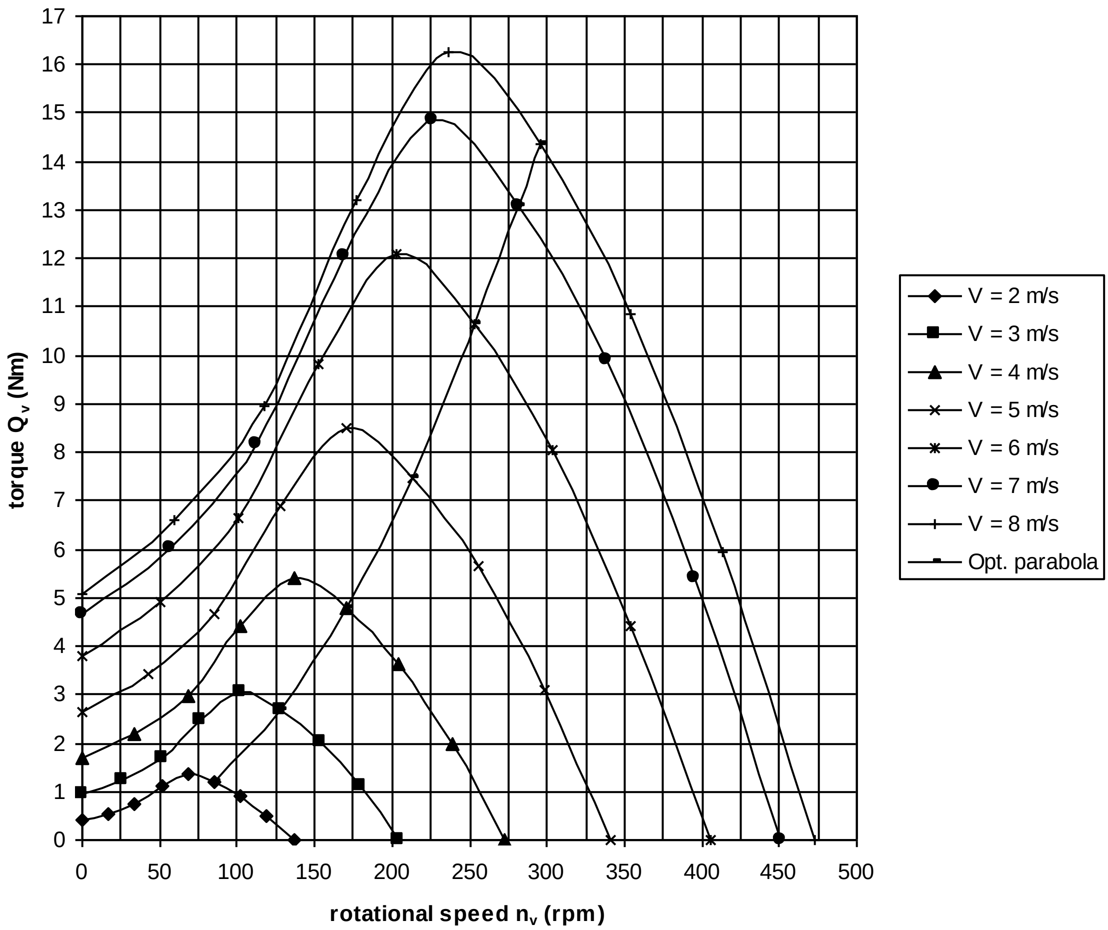

fig. 4 Q_v-n_v curves of the VIRYA-2.8B4 rotor for i = 2.5 and η_{tr} = 0.95

The optimum parabola which is going through the points with λ = 2.5, where C_p is maximum, is also drawn in figure 4.

8 Details of head and transmission

A composite drawing of the head and the transmission has been made in May 2012. Some details will be explained using this composite drawing.

The vane arm will be made of a 2 m long, 2” gas pipe. The rotor shaft will have a diameter of 30 mm and will be provided with tapered ends for connection of the rotor hub and the big 300 mm Polycord wheel. For the rotor shaft bearings, it is chosen to use the INA bearings type CR-B 30/83. This is a radial insert ball bearing with rubber interliner meant to be pressed in a pipe. For the head pipe a 350 mm long 3” gas pipe is chosen which is turned to an inside diameter of 82.6 mm at both ends. The rotor shaft is clamped in the bearings and therefore the positioning of the 300 mm Polycord wheel with respect to the 120 mm Polycord wheels can be adjusted.

The 2” pipe of the vane arm is not directly welded to the head pipe because this requires a complicated rounding of the pipe end at an angle of 45°. A 120 mm long piece of U-beam size 80 * 45 is welded to the 3” pipe and the 2” pipe is welded to this U-beam at a 45° angle.

The outer end of the 2” pipe is flattened and a strip size 70 * 8 * 750 mm is welded in this end. This strip makes a backwards angle of 23° with the pipe and so the angle in between the strip and the rotor axis is 22°. The aluminium vane blade size 2 * 750 * 750 mm is connected to this strip by means of three stainless steel 3” hinges. In between the hinges, two vane blade stops are connected to the strip and this stops prevents that the vane blade angle can become more that 87°. This prevents flutter of the vane blade at very high wind speeds.

A stainless steel pin is welded to the 2” pipe at a position such that the eccentricity is 295 mm. This pin has a diameter of 35 mm at the head bearings and also 35 mm at the hole in the pipe. The pin has a 47 mm collar below the pipe which is needed for the upper head bearing. The head turns in two INA Permiglide bearings. The upper bearing is flanged and has type PAF 35260 P11. The lower bearing is not flanged and has type PAP 3030 P11. The bearing housing is made of stainless steel and has the same dimensions as the bearing housing of the former VIRYA-3D windmill.

The head pin has a 20 mm hole inside. On the upper side, two sealed bearings size 12 * 28 * 8 mm are used. On the bottom side one sealed bearing size 12 * 28 * 8 mm is used. The 12 mm diameter vertical shaft turns in these bearings. The 120 mm Polycord wheel is clamped to the 12 mm shaft. The geometry of the wheel is chosen such that the hart of the V-groove coincides just at the centre of both upper bearings. By this construction, no bending moment is exerted to the vertical shaft by the pulling force in the Polycord string and this prevents bending of the shaft. The shaft is made of 12 h7 mm stainless steel shaft. This type of rod is normally supplied in a length of 3 mm, so the total shaft is built op from three ends which are connected to each other. This connection can be made by clamping blocks or by long nuts M12.

Both 120 mm Polycord wheels are made of massive aluminium rod. The big 300 mm aluminium Polycord wheel at the rotor shaft can be cast or can be built up from a rim, four or six spokes and a hub which are welded together. The V-groove is made in the rim after welding.

The 120 mm auxiliary Polycord wheel has two bearings size 12 * 28 * 8 mm inside. The shaft is a bolt M12 * 120 mm. This bolt is screwed in an aluminium block made of strip size 60 * 20 mm. This block is clamped around the head pin by means of two bolts M8 * 50. The head pin diameter is reduced to 34 mm at the block to realise the correct height of the auxiliary wheel. The correct angle of 45° of the wheel axis with respect to the rotor axis can be realised by turning the clamping block around the head pin.

The tower is identical to the tower of the former VIRYA-3D windmill. It has a 2 m long upper part made of 2” gas pipe and a 6 m long lower part made of angle iron and horizontal strips. The 2” pipe is clamped in the tower by means of four clamping blocks which are connected to the tower horizontals by threaded rods M10. The overlap in between the pipe and the lower tower part is about 0.5 m so the total tower height is about 7.5 m.

A bearing housing is welded at the bottom of the 2” gas pipe and a sealed bearing size 12 * 28 * 8 mm is used to support the vertical shaft at this position. Some extra bearings will be needed at positions of 1.5 m, 3 m and 4.5 m below this bearing. The bottom bearing will lie at a distance of about 1 m above the foundation. These bearings are mounted in a stainless steel bearing housing which has an extra sealing at the top to prevent entrance of water in the bearing. The bearing housing is connected to the horizontals of the tower by means of four stainless steel threaded rods M6.

For the transmission of the vertical shaft to the horizontal shaft of the rope pump several options are open. The distance in between the vertical shaft and the pump shaft can be chosen rather large so one may use a V-belt if the required reducing gear ratio is not very large. One may also use a 90° gear box or worm wheel box or another Polycord transmission. It has to be prevented that the vertical shaft is bent by this transmission.

It might be possible to connect a small PM-generator to the bottom of the vertical shaft and so the VIRYA-2.8B4 can be used for water pumping and battery charging at the same time if the wind regime is good enough to supply the required power. It has to be tested if the Polycord transmission is strong enough for the extra load of the generator.

9 References

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