Ideas about a piston pump with a floating valve for the VIRYA-3.6L2 windmill for irrigation of farmland from a river or a lake

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1 Introduction

Water is a basic need of mankind but many places on earth suffer from water shortage. Even if the average yearly rainfall is enough, there will be dry periods when water has to be pumped. Many places on earth have no access to the electricity grid and then one has to use hand pumps or pumps driven by a motor aggregate or by renewable energy like sun or wind. As fuel prices are rising, the use of sun or wind energy becomes more and more attractive in developing countries. But because of the required investment costs, the price of energy generated by sun or wind is much higher than the price of energy supplied by the grid. So if wind energy is used to power a pump, one has to use a windmill rotor with a high C_p and a pump with a high efficiency. Several ways how water can be pumped with a windmill are described in public report KD 490 (ref. 1).

The 4-bladed VIRYA-3.6L2 windmill was originally designed to drive a rotating positive displacement pump using a vertical shaft in the tower and an accelerating Polycord transmission in the windmill head. The original VIRYA-3.6L2 is described in public report KD 651 (ref. 2). The rotor diameter D = 3.6 m and the design tip speed ratio $\lambda_d = 2$.

However, to find a rotating positive displacement pump with the correct stroke volume in the market or to build one isn't simple. The Polycord transmission in the windmill head and the rotating vertical shaft in the tower centre also may give problems. So in this report KD 704 it is investigated if it is possible to use a single acting piston pump with a floating valve in combination with the VIRYA-3.6L2 rotor. A normal single acting piston pump is described in public report KD 294 (ref. 3). A single acting piston pump with a floating valve is described in public report KD 364 (ref. 4).

2 Description of the transmission to the piston pump with a floating valve

The CWD 2000 windmill was the first water pumping windmill which was equipped with a piston pump with a floating valve. It has been tested on the test field of the Eindhoven University of Technology for the period 1988 – 1990. The system worked nicely but there were some problems with the bearings of the rocker arm transmission which transforms the rotating movement of the crankshaft to the oscillating movement of the pump rod. The CWD 2000 has therefore never been used in CWD-projects, also because CWD was discontinued in July 1990 because the Ministry of Developing Countries has stopped financing.

The CWD 2000 was also the first water pumping windmill which was equipped with the hinged side vane safety system. This safety system is described, for blades with a cambered airfoil, in public report KD 223 (ref. 5). The VIRYA-3.6L2 will also be equipped with this safety system which is also used in all other VIRYA windmills.

For the original VIRYA-3.6L2 with a Polycord transmission, the eccentricity e in between the rotor shaft and the tower axis is chosen rather high (e = 0.369 m) because the vertical shaft gives a reaction moment which is working opposite the direction of the moment of the rotor thrust which turns the head out of the wind. If a piston pump is used, there is no reaction moment and therefore the eccentricity e can be chosen smaller for the same vane geometry. It is chosen that e = 0.3 m and this gives a ratio e / D = 0.3 / 3.6 = 0.0833 = 8.33 %. This is certainly enough to realise a stable safety system as a rotor with a design tip speed ratio $\lambda_d = 2$ has almost no self orientating moment. Another difference with the original VIRYA-3.6L2 is that now the eccentricity e can be chosen at the right side of the tower axis like it is also done for all other VIRYA windmills.

The CWD 2000 had an eccentricity e = 0.3 m for a rotor diameter D = 2 m. So e / D = 0.15 which is much higher than for the VIRYA-3.6L2. The reason for the very large eccentricity of the CWD 2000 has to do with the rocker arm transmission of this windmill. The CWD 2000 rocker arm has a length of 0.3 m and the turning point lies in the middle of the rocker arm, so at a distance of 0.15 m from the rotor axis and the pump rod axis.

A different construction will be chosen for the rocker arm of the VIRYA-3.6L2. For this windmill it is chosen that the length of the rocker arm l_r is twice the eccentricity e. So l_r = 0.6 m. This means that the stroke of the pump rod s_p is twice the stroke of the top of the connecting rod s_c. The main advantage of this construction is that the hinge in between the pump rod and the rocker arm makes only a small side wards movement and this limits the diameter of the required central hole in the head frame through which the pump rod moves.

The rotor shaft has two tapered ends. The rotor hub is clamped at the front shaft end by one central bolt M16. The crank is clamped at the back shaft end also by one central bolt M16. The crank is positioned such that a line perpendicular to the rotor shaft and lying at the middle of the crank bearing is intersecting with the tower axis. The rocker arm is positioned above the rotor axis which means that the connecting rod is pointing upwards.

The crank bearing is a heavy rubber sealed ball bearing. The rocker arm is swinging around a turning point using sealed needle bearings. The bearings in between the connecting rod and the rocker arm and in between the pump rod and the rocker arm are also sealed needle bearings.

The pump rod, the piston, the connecting rod and the rocker arm have a certain weight. This weight will make that the piston will always be at the lowest position if there is no wind. It is possible to balance this weight if the rocker arm is extended to the other side of the turning point and provided with a balancing weight. However, this balancing weight may give resonance problems at high rotational speeds. So it is assumed that all moving components of the transmission can be made that light than the weight isn't resulting in a too high starting wind speed. A sketch of the rocker arm transmission is given in figure 1.

The pump rod in between the rocker arm and the piston of the pump is made out of one pipe. This pipe rotates with the head if the wind direction changes. This may require a swivel in the pump rod but no swivel is needed if the piston can rotate easily enough in the cylinder.

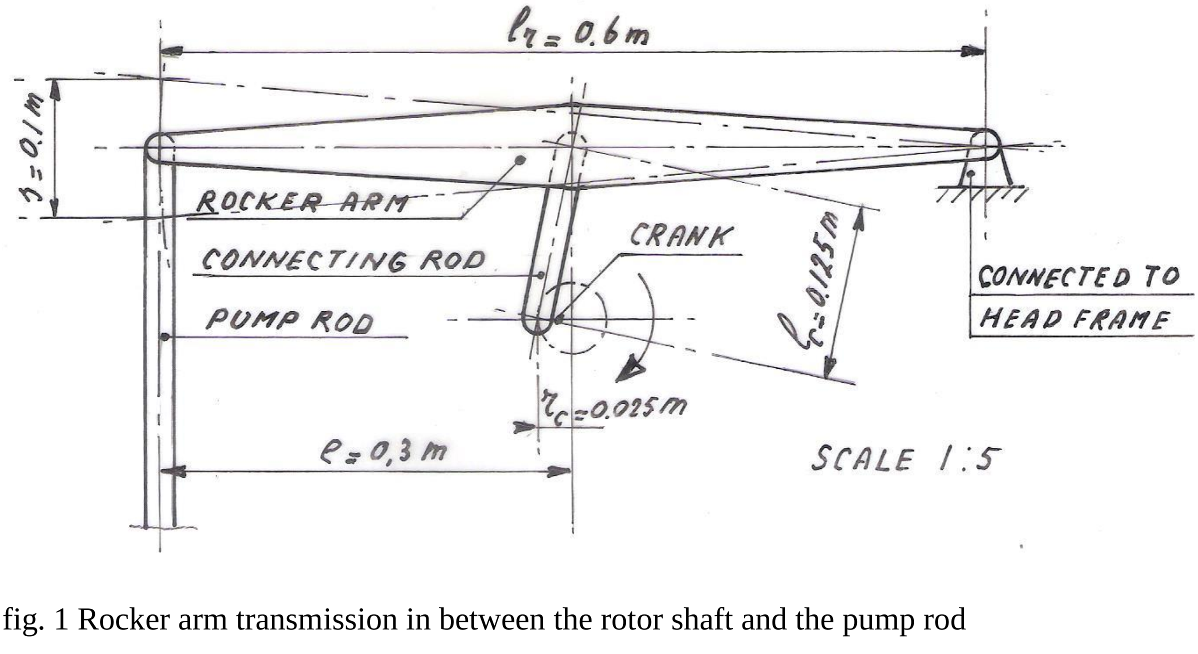

fig. 1 Rocker arm transmission in between the rotor shaft and the pump rod

3 Description of the pump

The pump is mounted just above the foundation at the heart of the tower. A sketch of the CWD 2000 pump is given in figure 2 of report KD 364. This pump has a floating piston valve which consists of two polypropylene hollow halves which are melted together. A very light valve with a high buoyancy force is created for a hollow valve. However, manufacture of a hollow valve is a rather tricky procedure.

It is also possible to use a massive polypropylene piston valve as the density of polypropylene ρ_p is smaller than water (ρ_p = 0.91 * 10^3 kg/m^3). The buoyancy force of a massive valve of a certain volume is much smaller than that of a hollow valve of the same volume but it can be large enough if the valve height is chosen large enough. Report R1009D of June 1989 (ref. 6) describes tests with different valve materials and valve constructions. I have a copy of this report but it is no longer available. Test on the TUE 5001 test rig with a massive polypropylene valve with a height of 60 mm are described in chapter 3.7 of R1009D. The valve closes already at a rotational speed of about 0.2 revolutions per second, so at about 12 rpm. A massive valve hasn’t been tested in combination with the CWD 2000 pump but I think that a massive polypropylene valve can be used for the VIRYA-3.6L2 pump. The starting behaviour with such a valve is now discussed.

The piston is in the lowest position if there is no wind because of the weight of the pump rod, the piston, the rocker arm and the connecting rod. The rotor torque increases sinusoidal if this weight is lifted and has a maximum value half way the upwards stroke, so for the about horizontal position of the crank. Assume that the oscillating components can be made that light that the required peak torque can just be supplied at a wind speed of 3 m/s. The design wind speed is also chosen 3 m/s. The design wind speed is the wind speed for which the rotor runs at its design tip speed ratio λ_d = 2.

The Q-n curves of the rotor for different wind speeds are given in figure 5 of KD 651 (ref. 2). This figure is copied as figure 2.

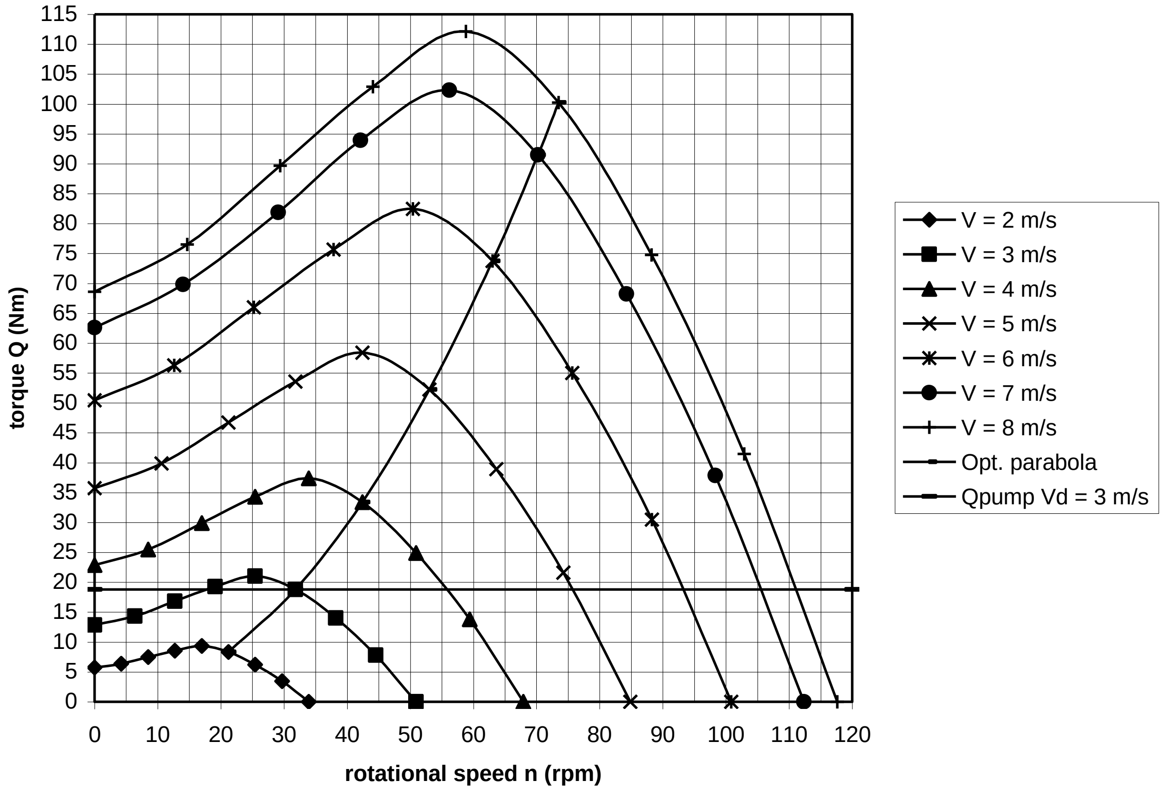

fig. 2 Q-n curves of the VIRYA-3.6L2 rotor for different wind speeds and optimum parabola

Figure 2 also contains a horizontal line for Q = 18.8 Nm which intersects with the optimum parabola at a wind speed of 3 m/s. This line is the constant torque for a rotating positive displacement pump but the average torque for a single acting piston pump. As the peak torque of a normal single acting piston pump is a factor π higher than the average torque, the peak torque will be π * 18.8 = 59 Nm. In figure 2 it can be seen that this torque requires a wind speed of almost 7 m/s at stand still position of the rotor if the pump has no floating valve and if the design wind speed is 3 m/s. So this starting wind speed is much too high and therefore a floating valve is needed.

So if the oscillating weights are light enough, the piston is lifted up to half way the upwards stroke, so for a 90° angle of rotation α of the rotor shaft from the lower dead centre, if the wind speed is 3 m/s. The movement of the piston from be lower dead centre up to half way the upwards stroke will be that slow at a wind speed of 3 m/s, that even a massive polypropylene piston valve will float. So the required torque is only needed to lift the weight. The required torque decreases sinusoidal if the piston is lifted more and is zero at the upper centre.

For the following downwards stroke, the weight will give a positive torque which also varies sinusoidal and this torque must be added to the rotor torque Q at a wind speed of 3 m/s! So the rotor accelerates and will have a rather large rotational speed when the piston arrives again at the lower dead centre. During the second upwards stroke, the piston valve may close at a certain point if the water speed in between the valve and the valve seat becomes high enough. But I expect that the rotor works enough as a flywheel to reach the upper dead centre for the second time without stopping somewhere during the upwards stroke. Practical tests on a prototype must prove if a massive polypropylene floating valve has enough buoyancy force to make that the rotor starts at least at the design wind speed V_d.

The piston construction is about the same as given in figure 2 of KD 364 except that the valve is massive. The thickness of the piston valve is about the same as the valve outer diameter. The piston valve has a rubber stop at the top. The leather cup is clamped in between two brass or stainless steel disks in which six holes are made. The foot valve is a thin disk but can also be made of polypropylene.

The pump is meant for low head irrigation and the water is pumped from a river or from a lake. The water is pumped in a reservoir and it is assumed that the outlet opening of the pressure pipe is lying 6 m higher than the water level of the river. So the static water height H = 6 m. Irrigation is done by gravity from the reservoir. It is assumed that the windmill is placed at a certain distance from the river shore and that the suction height is about 2 m. So the pressure height is about 4 m for a total static height H = 6 m.

The distance in between the river and the reservoir may be about 100 m so this means that the suction and the pressure pipes will be rather long. The flow supplied by a single acting piston pump is very fluctuating and as the water masses in the suction and the pressure pipes will be large for long pipes, this would result in high dynamic forces if no elastic element is used in the pump. This problem is normally solved by using air chambers.

The disadvantage of an air chamber in the pressure pipe is that the air will slowly dissolve into the water. Therefore an open air chamber will be used at the pressure side of the pump. An open air chamber is simply a concentric pipe around the pump rod. This pipe ends as high as possible, so just below the lower head bearing. The average water level in the pipe is the same as the water level at the outlet opening of the pressure pipe (if the pressure drop due to friction losses in the pressure pipe is neglected). The pressure drop due to friction losses can only be neglected if the pressure pipe has a relatively large inside diameter.

The fluctuation of the water speed in the suction pipe is flattened by a suction air chamber which is simply a vertical closed piece of pipe mounted close to the pump on top of the suction pipe. The volume of the suction air chamber must be about five times the stroke volume of the pump to get an almost constant flow in the suction pipe.

4 Calculation of the stroke volume, the flow and the pump geometry

It is assumed that the floating valve closes at a certain critical rotational speed n_{crit} which will be rather low for a massive polypropylene valve. If this critical rotational speed is reached, the valve will close half way the upwards stroke because there the speed is maximal. So the average torque for this rotational speed will be half the average torque for a valve which closes at the lower dead centre. The formula with which the average torque can be calculated for higher values than n_{crit} is derived in chapter 4 of KD 364. The ratio in between the average torque for a floating valve $Q_{av\ fv}$ and the average torque for a valve which closes at the lower dead centre Q_{av} is given in figure 4 of KD 364. This figure is copied as figure 3.

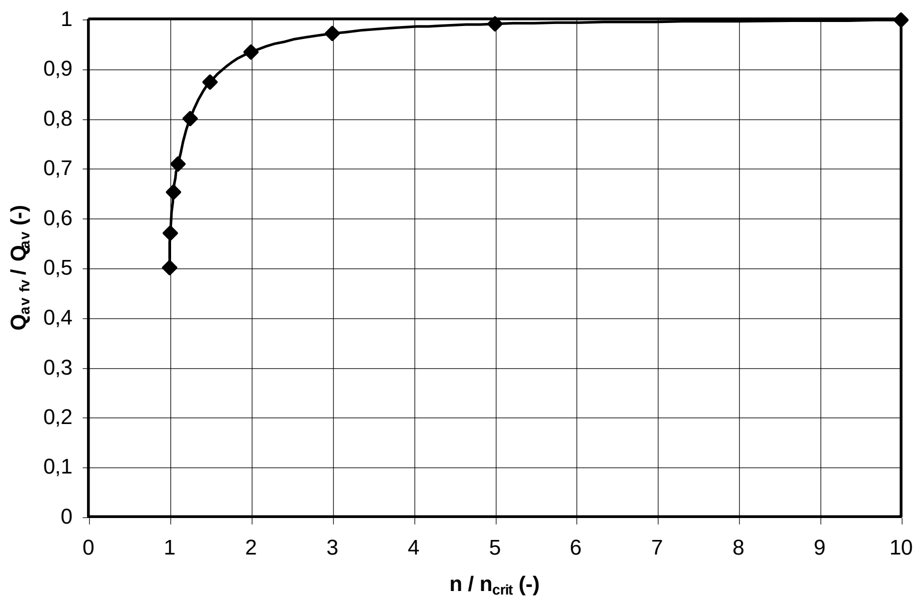

fig. 3 Values of $Q_{av\ fv}$ / Q_{av} as a function of n / n_{crit}

In figure 3 it can be seen that this ratio $Q_{av\ fv}$ / Q_{av} increases fast for values of n above n_{crit} and that it is 0.99 for n / n_{crit} = 5. Figure 3 is also valid for the volumetric efficiency η_{vol} (-).

The rotational speed of the VIRYA-3.6L2 rotor will only be very low during starting but at normal use, the rotational speed will be that high that the flow and the torque will be almost the same as for a pump without a floating valve. So for calculation of the stroke volume, the flow and the pump geometry, one can use the formulas for a normal single acting piston pump as given in report KD 294 (ref. 3).

The stroke of the piston s is the same as the stroke of the pump rod s_p and this stroke is twice the stroke of the top of the connecting rod s_c if the length of the rocker arm l_r is twice the eccentricity e. The theoretical stroke volume ∇ is given by formula 10 of KD 294. This formula is copied as:

$$\nabla_{th} = \pi/4 * D_p^2 * s \qquad (m^3) \qquad (1)$$

For the real stroke volume ∇_{real}, the volumetric efficiency η_{vol} (-) has to be taken into account.

$$\text{So } \nabla_{real} = \eta_{vol} * \pi/4 * D_p^2 * s \qquad (m^3) \qquad (2)$$

In these formulas, D_p is the piston diameter (m) and s is the piston stroke (m).

The flow q (m^3/s) is given by formula 15 of KD 294. This formula is copied as:

$$q = \eta_{vol} * \pi/4 * D_p^2 * s * n / (i * 60) \quad (m^3/s) \qquad (3)$$

In this formula n is the rotational speed (rpm) and i is the gear ratio. For the VIRYA-3.6L2, the crank shaft is directly driven by the rotor shaft, so i = 1.

The formula which links the rotor to the pump for the design wind speed V_d is given as formula 20 of KD 294. This formula is copied as:

$$4 * i * \eta_{tr} * \eta_p * C_{p\,max} * \rho V_d^2 * \pi R^3 = \rho_w * g * H * \eta_{vol} * D_p^2 * s * \lambda_d \qquad (4)$$

In this formula, i = the gear ratio (-), η_{tr} is the transmission efficiency (-), η_p is the pump efficiency (-), $C_{p\,max}$ is the maximum power coefficient of the rotor (-), ρ is the air density (kg/m^3), V_d is the design wind speed (m/s), R is the rotor radius (m), ρ_w is the water density (kg/m^3), g is the acceleration of gravity (m/s^2), H is the static water height (m), η_{vol} is the volumetric efficiency (-), D_p is the piston diameter (m), s is the piston stroke (m) and λ_d is the design tip speed ratio of the rotor (-).

Formula 4 can be written in many different ways depending on which factors one wants to have at the left and at the right side of the = sign. Assume one wants to calculate the piston diameter D_p for certain conditions. Formula 4 is now written as:

$$D_p = \sqrt{\{4 * i * \eta_{tr} * \eta_p * C_{p\,max} * \rho V_d^2 * \pi R^3 / (\rho_w * g * H * \eta_{vol} * s * \lambda_d)\}} \quad (m) \qquad (5)$$

Assume i = 1, assume η_{tr} = 0.98, assume η_p = 0.9, assume $C_{p\,max}$ = 0.38, assume ρ = 1.2 kg/m^3, assume V_d = 3 m/s, assume R = 1.8 m, assume ρ_w = 1000 kg/m^3, assume g = 9.81 m/s^2, assume H = 6 m, assume η_{vol} = 0.99, assume s = 0.1 m, assume λ_d = 2. Substitution of these values in formula 5 gives that D_p = 0.151 m = 151 mm. One will find another value for D_p if another stroke s is chosen.
Substitution of D_p = 0.151 m and s = 0.1 m in formula 1 gives ∇_{th} = 0.00179 m^3.
In figure 2 and in table 2 of report KD 651 it can be seen that the design rotational speed n_d at a design wind speed V_d = 3 m/s is 31.83 rpm. Substitution of η_{vol} = 0.99 (-), D_p = 0.151 m, s = 0.1 m, n = n_d = 31.83 rpm and i = 1 in formula 3 gives that q_d = 0.000941 m^3/s = 0.941 litres/s = 3.386 m^3/hour = 81.26 m^3/day. This is a substantial flow for irrigation at a wind speed of only 3 m/s.

The flow at higher wind speeds than 3 m/s will be substantial higher. Figure 2 shows that the point of intersection of the horizontal line for Q = 18.8 Nm intersects with the Q-n curves of the rotor for higher wind speeds at much higher rotational speeds. The rotational speed at a wind speed of 8 m/s (and higher) is about 112 rpm so a factor 112 / 31.83 = 3.519 higher than at V_d = 3 m/s. This means that the flow at V = 8 m/s is also a factor 3.519 higher than at V_d = 3 m/s. So at a wind speed of 8 m/s, the flow is 3.519 * 81.26 = 286 m^3/day. So if the real average wind speed is only somewhat higher than 3 m/s, the daily output will be much higher than 81.26 m^3/day.

5 Calculation of the maximum piston acceleration

It was assumed that the pump stroke s = 0.1 m and that the stroke of the top of the connecting rod s_c = 0.05 m for the chosen length l_r of the rocker arm. This means that the radius of the crank r_c = 0.025 m. It is assumed that the length of the connecting rod l_c is five times r_c so l_c = 0.125 m. The length of the rocker arm l_r is very long with respect to r_c, so the top of the connecting rod is moving almost along a straight line. For this situation, the movement of the top of the connecting rod is almost sinusoidal. The small deviation from a pure sinusoidal movement is neglected. The rotational angle α is taken in radians (rad). It is assumed that the rotational angle of the crank shaft α is zero for the crank in the horizontal position if the piston is half way the upwards stroke. The upwards movement from this point is taken positive. The stroke s_c at top of the connecting rod is then given by:

$$s_c = r_c \sin \alpha \quad \text{(m)} \tag{6}$$

Substitution of α = ½ π rad = 90° (upper dead centre) in formula 6 gives $s_c = r_c$. Substitution of α = 1 ½ π rad = 270° in formula 6 gives $s_c = -r_c$. So the total stroke in between the upper and the lower dead centre is 2 * r_c. The relation in between the angle α (rad), the angular velocity ω (rad/s) and the time t (s) is given by:

$$\alpha = \omega * t \quad \text{(rad)} \tag{7}$$

(6) + (7) gives:

$$s_c = r_c \sin (\omega * t) \quad \text{(m)} \tag{8}$$

The velocity at the top of the connecting rod V_c (m/s) is given by the first derivative.

$$V_c = f'(t) = \omega * r_c \cos (\omega * t) \quad \text{(m/s)} \tag{9}$$

The acceleration at the top of the connecting rod a_c (m/s^2) is given by the second derivative.

$$a_c = f''(t) = - \omega^2 r_c \sin (\omega * t) \quad \text{(m/s}^2\text{)} \tag{10}$$

(7) + (10) gives:

$$a_c = - \omega^2 r_c \sin \alpha \quad \text{(m/s}^2\text{)} \tag{11}$$

The relation in between the rotational speed n (rpm) and the angular velocity ω (rad/s) is given by:

$$\omega = \pi * n / 30 \quad \text{(rad/s)} \tag{12}$$

In figure 2 it can be seen that the maximum rotational speed at V = 8 m/s is about 112 rpm. Substitution of this value in formula 12 gives that ω = 11.73 rad/s. The acceleration is maximal for the upper and the lower dead centre so for α = ½ π rad and for α = 1 ½ π rad. Assume we take the lower dead centre, so α = 1 ½ π rad. This gives sin α = -1. Substitution of ω = 11.73 rad/s r_c = 0.025 m and sin α = -1 in formula 11 gives that a_c = 3.44 m/s^2. For the upper dead centre, so for α = ½ π rad, we find that a_c = -3.44 m/s^2.

Up to now we have given the formulas for s, V and a for the top of the connecting rod. The stroke, the velocity and the acceleration at the pump rod will have the double values because the length l_r of the rocker arm is twice the eccentricity e. So the maximum acceleration of the piston at the upper dead centre $a_p = 2 * a_c = 2 * -3.44 = -6.88$ m/s^2. So at the upper dead centre $g + a_c = 9.81 – 6.88 = 2.93$ m/s^2 which is still positive. This means that even at the upper dead centre, there is still a small pulling force acting on the pump rod.

However, during the downwards stroke there will be a certain pressure drop over the piston due to the hydraulic losses when the water flows through the holes in the piston disks and through the gap in between the piston valve and the valve seat. This pressure drop will be maximal for the highest downwards speed of the piston, so for $\alpha = \pi$ rad. But the acceleration force is just zero for this angle α and so the peak in both forces don't coincide. But the pressure drop over the piston gives a pushing force in the pump rod which may be larger than the pump rod weight. In this case, the pump rod is loaded by a small pushing force. So the pump rod must be made that stiff that it isn't sensible for buckling by this force.

6 Alternative transmission without a rocker arm

6.1 General

The transmission with a rocker arm as given in chapter 2 has as advantage that the horizontal movement of the top of the pup rod is very small. This allows a small hole in the head frame through which the pump rod passes. However, a rocker arm also has certain disadvantages.

The first disadvantage is that apart from the crank bearing, also bearings are needed at the turning point of the rocker arm, at the turning point at the top of the connecting rod and at the turning point at the top of the pump rod. These bearings have given problems for the CWD 2000 rocker arm because the angular movement is only small and the bearing is always loaded at the same place making that the grease is pressed away. Therefore sealed needle bearing were chosen for the VIRYA-3.6L2 rocker arm but this is a rather expensive solution.

The second disadvantage is that the rocker arm will be rather heavy if it is strong enough and it can be questioned if the weight can be made light enough to realize a sufficient low starting wind speed for a non balanced rocker arm.

The third disadvantage is that the rocker arm hinge lies at a distance of 0.6 m from the tower centre and so a rather long supporting structure has to be made to create a bearing at that place.

These problems are cancelled if the pump rod is directly connected to the crank. This option has been tested for the CWD 2740 windmill in about 1980 and it worked nicely. However, this option also has certain disadvantages.

The first disadvantage is that now the horizontal movement of the top of the pump rod is very large. So the head bearing must have a very large inside diameter.

The second disadvantage is that the open air chamber which is used to prevent large shock forces in the pressure pipe must also have a large inside diameter to prevent that the pump rod touches it.

The third disadvantage is that the rotor axis can have no eccentricity. This means that now an auxiliary vane has to be used to push the rotor out of the wind at high wind speeds.

The fourth problem is that the pump rod makes a small angle with the cylinder axis if the crank is horizontal. If the pump rod has a length of 6 m and if the crank has a length of 0.05 m, the maximum angle in between the axis of the pump rod and the axis of the cylinder is about a half degree. The leather cup of the piston is flexible enough to take this small wobbling movement of the piston so this isn't a real problem.

I think that the three other problems can be solved and that cancelling of the rocker arm makes the transmission much simpler and that maintenance will be reduced a lot.

6.2 Choosing a head bearing with a large inside diameter

The CWD 2740 had a stroke at the crank of 0.06 m but a pump with a rather large piston diameter to get a sufficiently large stroke volume. The problem of the rather large horizontal movement of the top of the pump rod was solved by using a 5” head pipe. However, this made that rather big bearings were needed at the top and at the bottom of the head pipe. It appeared to be difficult to seal these bearings and dust and water could entre easily which made it necessary to oil the bearing regularly. So this construction isn’t chosen for the VIRYA-3.6L2.

For one of my first windmills, the DRIEKA-4 and for the water pumping windmill CWD 5000 HW a big diameter ball bearing was used which is normally used to make that the front wheel shaft of a trailer can turn. The type which we used had a Z-shape cross section and the diameter of the upper flange is therefore much smaller than the diameter of the lower flange. These bearings are still available but are rather expensive (more than € 200 for the smallest type with a 400 mm lower flange).

At this moment also bearings are available with a U-shaped cross section and for this type the upper and the lower flange have about the same diameter. These types are much cheaper for the same load so this type is chosen. Information can be found on the Dutch website: www.middelbos.nl/agrarische-wagenbouw/draaikransen. I have chosen the type which can have an axial load of 750 kg with Art. Nr. 130011. The outer diameter of the upper flange A = 400 mm. The outer diameter of the lower flange B = 415 mm. The inner diameter of the upper flange C = 315 mm. The inner diameter of the lower flange D = 340 mm. The flange thickness G = 8 mm. The total height H = 55 mm. The price is € 77.70.

The minimum inside diameter is 315 mm, so there is plenty of space for the horizontal movement of the top of the pump rod. The maximum axial load is a mass of 750 kg so a force of about 7500 N. The mass of the head will be about 150 kg so the weight of the head is about 1500 N. This means that there is a large reserve for the extra load of the pulling force in the pump rod F_p. This force can easily be calculated with the formula:

$$F_p = \pi/4 \, D_p^2 * \rho_w * g * H \qquad (N) \qquad (13)$$

Substitution of D_p = 0.151 m, ρ_w = 1000 kg/m^3, g = 9.81 m/s^2 and H = 6 m in formula 13 gives that F_p = 1054 N. This is the static force to lift the water column. There will also some dynamic peak force when the piston valve closes. Assume that the maximum total force is 2000 N. Adding of 1500 N of the weight of the head gives a total maximum axial load of 3500 N. The maximum allowable load for the bearing is 7500 N so the bearing is strong enough.

The piston had a calculated diameter of 151 mm. So a cylinder has to be found with about this inside diameter. Assume that the cylinder is made out of PVC pipe. A very standard value has an outside diameter of 160 mm and a wall thickness of 4 mm and so the inside diameter is 152 mm. This seams a good choice. Assume that this pipe also used for the open pressure air chamber. So this air chamber also has an inside diameter of 152 mm. The side wards movement of the heart of the pump rod is 100 mm for a stroke of 100 mm. This means that the pump rod pipe can have a maximum outside diameter of about 50 mm. It seems possible to make a pump rod from aluminium pipe with this outside diameter which is light and stiff enough.

It is assumed that no swivel is used in the pump rod. This means that the piston will rotate in the cylinder if the head turns because of change of the wind direction. The leather cup will have some friction in the cylinder so the bearing at the top of the pump rod must be able to supply a certain twisting moment.

6.3 Choosing a safety system for a non eccentric rotor

If the rotor axis has no eccentricity with respect to the tower axis, the rotor thrust will give no moment which pushes the head out of the wind. So an auxiliary vane is needed to supply this moment. The use of an auxiliary vane in stead of an eccentrically place rotor is explained in chapter 4 of report KD 485 (ref. 7).

In figure 16 of KD 485 a cylindrical vane and a flat vane with an aspect ratio 2 : 1 are compared with an eccentrically placed rotor with a geometry such that the moment for a yaw angle $\delta = 0°$ is the same. It can be seen that the moment of a flat vane is much higher than for an eccentrically place rotor. At an angle of about 75° there is a strong discontinuity because the vane changes from a drag body into a lift body. This means that he rotor will turn out of the wind with a much larger yaw angle than necessary. The moment for a cylindrical vane is also larger than for an eccentrically place rotor but the difference is much smaller and there is no discontinuity as a cylindrical vane works only as a drag body. So a cylindrical vane is chosen. For the main vane system it is chosen to use the hinged side vane safety system as described in chapter 3.2 of KD 485 and in report KD 223 (ref. 5).

7 Conclusions

To my opinion, use of the VIRYA-3.6L2 in combination with a piston pump with a massive floating valve is a very realistic option if a large water flow is needed for irrigation of farmland from a river or a lake. It seems better to use a cylindrical auxiliary vane and a pump rod which is directly connected to the crank than an eccentrically placed rotor with a rocker arm. Real use of this windmill requires making of a composite drawing, detailed drawings, a manual and manufacture and testing of a prototype. I won’t do this but I will support skilled people who want to do this professionally, as much as possible.

8 References

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