Ideas about a diaphragm pump with three in line diaphragms driven by a 0.37 kW asynchronous 4-pole, 3-phase motor or a permanent magnet DC motor frame size 71 and a reducing 2-step gear box frame size 32 of manufacture Rossi

ing. A. Kragten

January 2013

KD 542

Engineering office Kragten Design
Populierenlaan 51
5492 SG Sint-Oedenrode
The Netherlands
telephone: +31 413 475770
e-mail: info@kdwindturbines.nl
website: www.kdwindturbines.nl

Contains page

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 sun or wind energy is used to power an electric pump, one has to use a pump with a high overall efficiency. This means that the ratio in between the obtained hydraulic energy in the water and the required electrical energy has to be as high as possible. However, a high efficiency should be realised at the lowest possible investment costs for the pump and the pump motor. Local manufacture of the pump may result in a rather low pump price.

The overall efficiency of an electric pump depends on the efficiency of the pump motor, the efficiency of the pump, the efficiency of the transmission in between motor and pump, the losses in the cables from the energy source to the pump motor and the hydraulic losses in the piping in between the well and the pump and in between the pump and the reservoir in which the water is pumped.

The highest motor efficiency is realised for a permanent magnet DC motor. An efficiency of 0.8 is possible for a small motor. However, these motors are rather expensive and may be difficult to obtain in developing countries. The nominal rotational speed is also higher than for a 4-pole asynchronous motor so a larger reducing gearing will be needed. The cable losses will be rather high as the nominal voltage is only 24 V. An alternative is to use an asynchronous 4-pole, 3-phase motor which runs at a nominal speed of about 1400 rpm. The motor efficiency η_m of such a motor with frame size 71 is about 0.65. The cable losses are low because a high 230/400 V, 3-phase voltage is chosen. It is estimated that the cable efficiency η_c for normal currents is 0.98. The highest pump efficiency η_p is possible for a piston pump. An efficiency of maximum 0.9 has been measured for single acting piston pumps. It is expected that a well designed diaphragm pump will have an efficiency of 0.88. The highest transmission efficiency is realised if there is no reducing transmission and if ball bearings are used at the crank shaft. However, if a direct drive transmission is used, the crank shaft of the pump will run at a very high rotational speed which causes severe acceleration problems. Therefore a reducing gearing is chosen using a 2-step gear box of manufacture Rossi. The membranes will be driven by a crank shaft with ball bearings and the total efficiency of the gear box and the crank transmission is expected to be 0.92. So the maximum total efficiency η_{tot} of the pump and an asynchronous motor including cables $\eta_{tot} = \eta_c * \eta_m * \eta_t * \eta_p = 0.98 * 0.65 * 0.92 * 0.88 = 0.52$. The hydraulic losses in the piping are minimised if the flow through the pipe is not fluctuating and if smooth pipes with sufficient inner diameter are used. The pipe resistance results in an extra dynamic head.

The main disadvantage of a single acting piston pump which is used in traditional water pumping windmills is that the torque and the flow vary sinusoidal during the upwards stroke and that they are zero during the downwards stroke. The peak torque is a factor π times the average torque. If an elastic element with a volume variation of about five times the stroke volume of the pump is added just after the pump, the flow coming out of this elastic element will be almost constant. This principle is used in the Solaflux solar pump which was tested in combination with the VIRYA-3B3 windmill already in 2003. This combination is described in chapter 5 of public report KD 490 (ref. 1).

Already in 1993 I have designed and drawn a diaphragm pump with three diaphragms driven by a permanent magnet DC motor through a reducing tooth belt transmission. This pump is described in report KD 03 (ref. 2, in Dutch). An important advantage of using three diaphragms is that the flow is almost constant (see chapter 2).

Now, 20 years later, I have picked up this idea but I use a more elegant gear box transmission with a more powerful flanged motor. Advantages of a diaphragm pump are that it is not sensible to small sand particles in the water and that it is self priming. The pump is meant for an open well and will be mounted some meters above the water level. It is assumed that the total static head is 40 m.

2 Determination of the fluctuation of the flow in the pressure pipe

A diaphragm will suck water from the suction pipe during the upwards stroke and will press water in the pressure pipe during the downwards stroke. The pump will be mounted as close as possible to the water level in the well so the suction pipe will be rather short. A diaphragm pump can supply a rather high pressure so the pressure pipe will be long. The fluctuation of the flow in the pressure and the suction pipe will be identical but 180° out of phase. The highest torque is required during the downwards stroke.

At this moment we only look at the flow which is pressed in the pressure pipe during the downwards stroke. The flow of one diaphragm to the pressure pipe varies sinusoidal during the downwards stroke and is zero during the upwards stroke. The angle in between the eccentrics is 120° so there is an angle of 120° in between the flow curves of the three diaphragms. The flow q for a certain rotational speed of diaphragm U during the outwards stroke is given by:

$$q = C * \sin\alpha \qquad (m^3/s) \qquad (0° < \alpha < 180°) \qquad (1)$$

The coefficient C depends on the stroke, the diaphragm diameter and the rotational speed. The angle α is the rotational angle of the crank shaft measured from the bottom dead centre. The three diaphragms are labelled U, V and W. Diaphragm U is the diaphragm closest to the gear box. The total flow of the three diaphragms q_{tot} is given by:

$$q_{tot} = C \{\sin\alpha + \sin(\alpha - 120°) + \sin(\alpha - 240°)\} \qquad (m^3/s) \qquad (2)$$

The contribution of each diaphragm to q_{tot} has been calculated for values of α increasing with 15°. The result of the calculation is given in table 1. The coefficient C is chosen 1.

In the last column for all three diaphragms it can be seen that every 60° there is a small fluctuation with a maximum value of 1 and a minimum value of 0.86603. The average value of a half sinus over 360° is $1/\pi$ times the peak value. So the average value of three halve sinuses over 360° is $3/\pi = 0.95493$ times the peak value. The maximum positive fluctuation around the average value is about 4.5 %. The maximum negative fluctuation around the average value is about 9.3 %. These fluctuations are rather small and it is assumed that they can be flattened by a flexible 4 meter long suction pipe and by a pressure pipe which is flexible for the first 4 meters. The flow q of diaphragms U, V and W, the total flow of all three diaphragms q_{tot} and the average total flow $q_{tot\ av}$ is given in figure 1 for C = 1.

On a certain scale, the variation of the flow q is congruent to the variation of the torque Q. So figure 1 can also be seen as a Q-α graph. On a certain scale, the variation of the flow for one diaphragm is also congruent for the variation of the downwards speed of the diaphragm.

	q diaphragm U	q diaphragm V	q diaphragm W	q_{tot} all three diaphragms
α (°)	sinα	sin(α – 120°)	sin(α – 240°)	
0	0	0	0.86603	0.86603
15	0.25882	0	0.70711	0.96593
30	0.5	0	0.5	1
45	0.70711	0	0.25882	0.96593
60	0.86603	0	0	0.86603
75	0.96593	0	0	0.96593
90	1	0	0	1
105	0.96593	0	0	0.96593
120	0.86603	0	0	0.86603
135	0.70711	0.25882	0	0.96593
150	0.5	0.5	0	1
165	0.25882	0.70711	0	0.96593
180	0	0.86603	0	0.86603
195	0	0.96593	0	0.96593
210	0	1	0	1
225	0	0.96593	0	0.96593
240	0	0.86603	0	0.86603
255	0	0.70711	0.25882	0.96593
270	0	0.5	0.5	1
285	0	0.25882	0.70711	0.96593
300	0	0	0.86603	0.86603
315	0	0	0.96593	0.96593
330	0	0	1	1
345	0	0	0.96593	0.96593
360	0	0	0.86603	0.86603

Table 1 flow q for each diaphragm and flow q_{tot} for all three diaphragms for C = 1

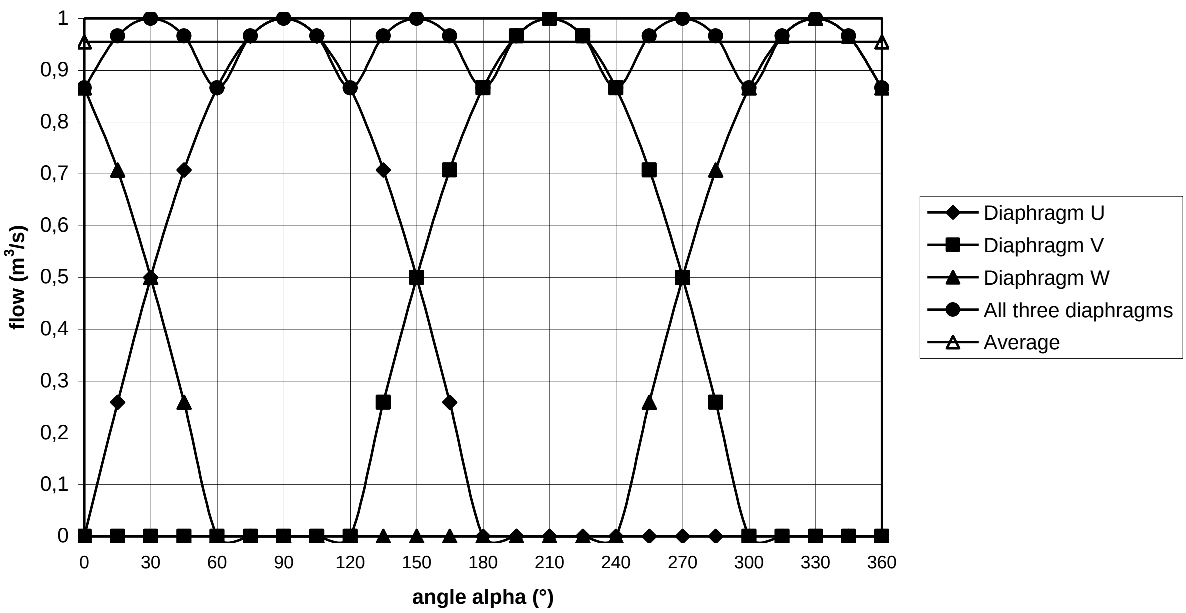

fig 1 fluctuation of the flow of each diaphragm q and of the total flow q_{tot} for C = 1.

3 Description of the motor, the transmission and the pump (see figure 2 page 14)

A 370 W asynchronous 4-pole, 3-phase motor is chosen. This motor has frame size 71 and runs at a nominal rotational speed of about n = 1400 rpm. This motor can be coupled to a 230/400 V, 3-phase grid. If only a 1-phase grid is available, one can use a 1-phase motor of the same power and frame size. A 1-phase motor in reality is a 2-phase motor and the second phase is created by a big capacitor. If a 3-phase or 1-phase grid isn't available and if the energy comes from a battery charging windmill or from a solar panel one has to use a 3-phase or 1-phase inverter.

Another option is to use a permanent magnet motor if it has the same frame size and power and about the same rotational speed. Permanent magnet DC motors with the same frame size as asynchronous motors are supplied by the Dutch company Creusen (www.Creusen.nl). Motors of this frame size are supplied for 24 V in different powers and nominal rotational speeds. Type 71L-2GP with P = 350 W and n = 1500 rpm seems the best option. If this motor is used in combination with a solar panel or a windmill, the real voltage will be higher than 24 V. It is assumed that a voltage controller is used which limits the voltage up to 28 V. This means that the maximum rotational speed is 1500 * 28 / 24 = 1750 rpm. This is a factor 1.25 higher than the rotational speed of a 4-pole asynchronous motor which means that the reducing gear ratio must be a factor 1.25 higher to get the same rotational speed of the crank shaft.

It might be possible to use a windmill which is directly supplying a 3-phase current. As the frequency of this current must be rather high, one has to use a multi pole generator if the generator is direct drive. A 22-pole generator of this kind is described in report KD 506 (ref. 3).

The advantage of an asynchronous motor is that the outside dimensions are standardised and that therefore any brand can be chosen. Motors of frame size 71 are normally supplied with a foot B3 but can also be supplied by a flange. A gear box is chosen which has a flange at the back side which means that the motor must have a flange too. The smallest gear box of Rossi has been chosen. This gear box has size 32. I have an old catalogue type E94 of Rossi in which coaxial gear motors are given. An on line catalogue can be found on the website of Rossi: www.rossi.com by following the path: products – coaxial gear motors Cat. E – pdf Coaxial gear reducers and gear motors Catalogue E04 (1.68 Mb). A picture of the gearbox plus motor is given on page 50 of the online catalogue. The rotational speeds, torques, safety factors and gear box and motor specifications for a 0.37 kW motor as a function of the gear ratio i are given on page 30 and 31 of the on line catalogue. A combination of a 2-steps gear box size 32 and a 4-pole motor size 71 is available for eight different gear ratios i being 5.06, 6.33, 7.29, 8.12, 9.57, 10.8, 13.5 and 16.5. The safety factor fs decreases at increasing gear ratio i and is smaller than 1 for the two largest gear ratios. This means that it isn't allowed to load the shaft higher than the nominal torque for these two largest gear ratios.

In the first instance it is chosen to use the gear ratio i = 5.06 for the 4-pole asynchronous motor and i = 6.33 for the permanent magnet DC motor. These ratios differ a factor 6.33 : 5.06 = 1.25.

The slow shaft of the gear box has a diameter of 16 mm and a length of 30 mm. It is provided with a 5 mm key which juts out 2 mm out of the shaft. The front bearing of the gear box is a double row ball bearing of probably size 17 * 40 * 16 mm. The dynamic load factor of this bearing C = 14800 N. The static load factor C_0 = 9000 N.

The crankshaft is made out of one 55 mm steel bar and has a diameter of 25 mm in between the cranks. The front side runs in a sealed ball bearing size 20 * 52 * 15 mm. For this bearing it is valid that C = 15900 N and C_0 = 7800 N. The back side is shifted over the gear box shaft and makes use of the gear box bearing. The crank shaft has three cranks which make an angle of 120° with respect to each other. A crank is realised by an eccentric.

The static imbalance force of one connecting rod and one eccentric is balanced by the static imbalance force of the other crank rods and eccentrics. However, because there is a distance of 100 mm in between the hart of the connecting rods, a certain dynamic imbalance moment will remain. It is expected that this dynamic imbalance moment is acceptable.

For the crank bearings it is chosen to use sealed ball bearings size 40 * 68 * 15 mm. So the outside diameter of the eccentric is 40 mm. The crank radius r or the eccentricity e is chosen 7 mm = 0.007 m which means that the stroke s of each connecting rod is 14 mm = 0.014 m. The bearing is axially fixated to the eccentric by retaining rings.

The load of the front connecting rod is almost completely taken by the front crank shaft bearing. The load of the central connecting road is taken about equal by the front crank shaft bearing and by the gear box bearing. The load of the back connecting rod is almost completely taken by the gear box bearing.

The connecting rod has to be made as light as possible to minimise imbalance forces. The connecting rod is therefore made out of aluminium. The thickness of the connecting rod is chosen the same as the width of the crank bearing, so 15 mm. The connecting rod is made from aluminium strip size 80 * 15 mm. Axial fixation of the bearing in the connecting rod is realised by a press fitting. The diaphragm is clamped in between two aluminium disks which have a diameter of 40 mm and a thickness of 3 mm. The disks are connected to the connecting rod by a central stainless steel tapered screw M8 * 25 and some locking liquid. The diaphragm housing is connected to the bottom side of an aluminium sheet which is bolted to the gear box foot which has a height of 75 mm. The aluminium sheet has a width of 150 mm and a thickness of 10 mm. This sheet has three 44 mm holes at the position of the diaphragms. So only a 2 mm wide ring of the diaphragm is visible from above and this reduces destruction of the rubber by ultra violet light.

A connecting rod has no cross head which means that the diaphragm disks are tilting a little in the diaphragm housing. The tilting angle depends on the length of the connecting rod and the eccentricity. The distance in between the hart of the gear box axis and the upper side of the diaphragm when it is in the middle position is 75 + 10 = 85 mm. The tilting angle is about 4.7° for this distance and for an eccentricity of 7 mm. This small tilting angle can easily be taken by a diaphragm. The outside of a diaphragm is clamped in between the aluminium strip and the central part of the diaphragm housing. Two tapered chambers with a diameter of 80 mm are made in the bottom side of the aluminium strip and in the upper side of the central part. So the average diaphragm diameter is D_{av} = (40 + 80) / 2 = 60 mm.

The three diaphragms are made out of one flat rubber sheet with a thickness of 3 mm. A stainless steel sheet with a thickness of 2 mm is mounted in between the central part and the lowest part of the diaphragm housing. The three suction and the three pressure valves are mounted against this stainless steel sheet which is provided with six 12 mm holes at a pitch circle of 32 mm for each valve. The suction and pressure valves are spring loaded. All valves are identical and have a diameter of 50 mm and a thickness of 3 mm. The valves are made out of strong plastic like PUM or polyoxmethyleen (supplied as Delrin, Ertacetal or Hostaform). The valve stroke is chosen 3 mm. The valve has a rubber stop at the open position. There are O-rings type ASR 144 around each valve at both sides of the stainless steel sheet to prevent water leakage. There are O-rings type ASR 237 at the upper side of each diaphragm to fixate the diaphragm. The central and the lower parts are made out of aluminium strip size 150 * 20 mm and size 150 * 40 mm but they can also be made of PUM or cast from aluminium for serial production.

At this moment it is chosen to position the suction valves at the left side and the pressure valves at the right side when one is looking to the front side of the gear box. The suction and the pressure opening are both chosen at the front side of the diaphragm housing.

A 1” hose pillar is screwed in both openings. A 4 m long reinforced hose with an inner diameter of about 25 mm is connected to the suction hose pillar. Another 4 m long identical hose is connected to the pressure hose pillar. These hoses are functioning as elastic element to flatten the little flow fluctuation which is explained in chapter 2. The remaining part of the pressure pipe can be made of non elastic plastic pipe. The pump is normally resting on the diaphragm housing. To prevent falling backwards, two aluminium studs are mounted at the backside. A sketch of the pump is given in figure 2 at page 14.

4 Calculation of the stroke volume, the flow, the power and the connecting rod force

The total stroke volume ∇ of three diaphragms is given by:

$$\nabla = 3 * \pi/4 * D_{av}^2 * s \qquad (m^3) \qquad (3)$$

The stroke volume will vary a little depending on the pressure because the diaphragm will bend to the inside during the suction stroke and bend to the outside during the pressure stroke but this effect is neglected at this moment.

The flow q is given by:

$$q = \nabla * n / 60 \qquad (m^3/s) \qquad (4)$$

(3) + (4) gives:

$$q = \pi * D_{av}^2 * s * n / 80 \qquad (m^3/s) \qquad (5)$$

The nominal rotational speed of the slow gear box shaft n = 277 rpm for a gear ratio i = 5.06 and a rotational speed of the motor of 1400 rpm. Substitution of D_{av} = 0.06 m, s = 0.014 m and n = 277 rpm in formula 5 gives that q = 0.000548 m^3/s. This is about 1.97 m^3/hour or about 47 m^3/day. This is a substantial amount of water which indicates that the pump can be used for drinking water and even for small irrigation. The hydraulic power P_{hyd} is given by:

$$P_{hyd} = \rho_w * g * H * q \qquad (W) \qquad (6)$$

In this formula H is the total head. H is the sum of the static head H_{stat} and the dynamic head H_{dyn}. The static head is the height in between the water level in the well and the height of the outlet opening of the pressure pipe. The dynamic head is caused by the pipe losses in the suction and the pressure pipe. The dynamic head depends on the flow, the length of the pipes and very much on the inside pipe diameter. Calculation of the dynamic head is out of the scope of this report. It is assumed that the dynamic head is 10 % of the static head. Assume H_{stat} = 40 m and H_{dyn} = 4 m so H = 44 m. Substitution of ρ_w = 1000 kg/m^3, g = 9.81 m/s^2, H = 44 m and q = 0.000548 m^3/s in formula 6 gives P_{hyd} = 237 W. The required mechanical power P at the pump shaft is given by:

$$P = P_{hyd} / (\eta_t * \eta_p) \qquad (7)$$

Substitution of P_{hyd} = 237 W, η_t = 0.92 and η_p = 0.88 in formula 7 gives P = 292 W. So a 370 W motor seems a good choice and has even some reserve if the efficiency is lower than the values as used for the calculation. The required electrical power P_{el} is 292 / (0.65 * 0.98) = 459 W.

Another aspect is the bearing load and the bending stress in the crank shaft due to the force in the connecting rods. For calculation of the connecting rod force it is assumed that the pump has a suction head H_s = 3 m and that total height H = 44 m. So the pressure head H_p = 41 m. The pressure drop Δp over the diaphragm during the downwards stroke is given by:

$$\Delta p = \rho_w * g * H_p \qquad (N/m^2) \qquad (8)$$

It is assumed that this pressure drop works over that part of the diaphragm within the average diameter. The average diaphragm area is given by:

$$A_{av} = \pi/4 * D_{av}^2 \qquad (m^2) \qquad (9)$$

The static force in the connecting rod F_{stat} is given by:

$$F_{stat} = \Delta p * A_{av} \qquad (N) \qquad (10)$$

(8) + (9) + (10) gives:

$$F_{stat} = \rho_w * g * H_p * \pi/4 * D_{av}^2 \qquad (N) \qquad (11)$$

Apart from the static load, there is also working a dynamic load in the pump rod which is caused by acceleration of the pump rod. As the stroke is rather small, this dynamic load is neglected at this moment. Substitution of ρ_w = 1000 kg/m^3, g = 9.81 m/s^2, H_p = 41 m and D_{av} = 0.06 m in formula 9 gives F_{stat} = 1137 N.

A crank bearing has size 40 * 68 * 15 mm with a dynamic load factor C = 16800 N and a static load factor C_0 = 9300. The dynamic load factor C is used for the calculation of the lifetime of the bearing. The lifetime L_h in hours is given by:

$$L_h = 10^6 * (C/P)^3 / (60 * n) \qquad (hours) \qquad (12)$$

Substitution of C = 16800 N, P = F_{stat} = 1137 N and n = 277 rpm in formula 12 gives that L_h = 194095 hour = 22 year. So the crank bearing is certainly strong enough.

The front crank shaft bearing has size 20 * 52 * 15 mm. This bearing has a dynamic load factor C = 15900 N and a static load factor C_0 = 7800 N. The gear box bearing is a double row ball bearing with probably size 17 * 40 * 17 mm. This bearing has a dynamic load factor C = 14800 N and a static load factor C_0 = 9000 N.

In figure 1 it can be seen that during 60°, only one connecting rod is having a pressure force but that during the next 60° there will be a pressure force in two connecting rods. A certain combination of two connecting rods exists during 1/6 of one revolution.

A combination of the central connecting rod V and the front connecting rod W gives the largest load on the front crank shaft bearing. A combination of the central connecting rod V and the back connecting rod U gives the largest load on the front gear box bearing. The load on these bearings depends on the distances in between the hart of the bearings. The distance in between the crank bearings is 100 mm. The distance in between the front crank shaft bearing and bearing of connecting rod W is 40 mm. It is assumed that the distance in between the gear box bearing and bearing U is 40 mm too. The reaction force on the front crank shaft bearing is called F_A. The reaction force on the gear box bearing is called F_B. The maximum value of F_A is found by taking the balance of moments around point B. It is found that $F_{A\,max}$ = (F_{stat} * 240 + F_{stat} * 140) / 280 = F_{stat} * 380 / 280 = 1.357 F_{stat}. This gives F_A = 1543 N for F_{stat} = 1137 N. Substitution of C = 15900 N, P = F_A = 1543 N and n = 277 rpm in formula 12 gives that L_h = 65836 hour = 7.5 year. This is acceptable so the front crank bearing is strong enough.

The maximum load on the gear box bearings F_B is the same as the load F_A. Substitution of C = 14800 N, P = F_A = 1543 N and n = 277 rpm in formula 12 gives that L_h = 53095 hour = 6.1 year. This is acceptable so the gear box bearing is also strong enough. The real lifetime may be longer than the calculated value because for normal lifetime calculations it is assumed that the load is working on the bearing during one full revolution. For the crank bearings the load is only working during a half revolution. For the crank shaft bearing and the gear box bearing, the maximum load is working only during 1/6 of a revolution.

The static force in a connection rods gives a bending moment in the crank shaft. The bending moment is maximal for the middle of the crank shaft. The bending moment is calculated for the situation that there is only a force acting on connection rod V. The force of connection rods U and W is neglected. So $F_A = F_B = ½ F_{stat}$ for this situation. The bending moment M is found by taking balance of moments around the middle of the crank shaft. This gives M = ½ F_{stat} * 140 = ½ * 1137 * 140 = 79590 Nmm. The bending stress σ in a shaft with a diameter d is given by:

$$\sigma = 32 M / (\pi * d^3) \qquad (N/mm^2) \qquad (13)$$

The shaft has a diameter of 25. Substitution of M = 79590 Nmm and d = 25 mm in formula 13 gives that σ = 52 N/mm^2. This is a low stress so the crank shaft is certainly strong enough. The crank shaft will bend because of the crank rod forces acting on it. Again only the crank rod force acting on crank rod V is taken into account. The deflection of a shaft f supported at both ends for a point load F in the centre is given by:

$$f = F * l^3 / (48 E * I) \qquad (mm) \qquad (14)$$

$$I = \pi/64 * d^4 \qquad (mm^4) \qquad (15)$$

(14) + (15) gives:

$$f = 4 * F * l^3 / (3 * \pi * E * d^4) \qquad (mm) \qquad (16)$$

Substitution of F = F_{stat} = 1137 N, l = 280 mm, E = 2.1 * 10^5 N/mm^2 and d = 25 mm in formula 16 gives that f = 0.13 mm. So the deflection of the crank shaft is only very little which means that the crank shaft is stiff enough.

5 Determination of the acceleration force in the connecting rod

The acceleration of the diaphragm is maximal at the bottom and top dead centre. The acceleration must be lower than the acceleration of gravity otherwise the water will cavitate when the under pressure below the diaphragm becomes lower than 1 bar. The acceleration of gravity g = 9.81 m/s^2 and the maximum suction head is about 10 m if the water isn't accelerated. Cavitation means that a bubble of vacuum is created which implodes at increasing pressure. This imploding gives large shocks forces in the system. The maximum acceleration force a_{max} is given by:

$$a_{max} = r * \omega^2 \qquad (N) \qquad (17)$$

ω is the angular velocity in radial per second. The relation in between the radial velocity and the rotational speed in rpm is given by:

$$\omega = \pi * n / 30 \qquad (rad/s) \qquad (18)$$

Substitution of n = 277 rpm in formula 18 gives that ω = 29 rad/s. Substitution of r = 0.07 m and ω = 29 rad/s gives that a_{max} = 5.9 m/s^2.

The maximum suction head for an acceleration of 5.9 m/s^2 is about 4 m which seems acceptable. The maximum dynamic force acting on the pump rod F_{dyn} is given by:

$$F_{dyn} = m * a_{max} \quad \text{(N)} \qquad (19)$$

The connecting rods are made as light as possible. It is estimated that the mass of a connecting rod, the part of the diaphragm within D_{av} and the clamping disks is 150 gram = 0.15 kg. Substitution of m = 0.15 kg and a_{max} = 5.9 m/s^2 in formula 19 gives that F_{dyn} = 0.9 N. In chapter 4 it was calculated that F_{stat} = 1137 N. So the dynamic force is only very low with respect to the static force. However, this is only true if there is a good working elastic element just behind the pump. The elastic element is formed by a 4 m long, 1" reinforced hose.

6 Determination of the water speed and the acceleration of the valves

If the pump is running at a rotational speed of 277 rpm, so at 4.6 revolutions per second, the valves have to open and close with a frequency of 4.6 Hz. This is rather fast and a valve will only be able to follow this frequency if it is very light and if the spring force which closes the valve is strong enough. The valves therefore have a stroke of only 3 mm and are made from a light plastic like PUM which has a density ρ_v of about 1.4 * 10^3 kg/m^3 or 1.4 * 10^{-6} kg/mm^3. A valve has a thickness of 3 mm, an outside diameter of 50 mm and an inside diameter of 10 mm. So the valve volume $\nabla = \pi/4\ (50^2 - 10^2) * 3 = 5655$ mm^3. So the valve mass m_v is given by m_v = 5655 * 1.4 * 10^{-6} = 7.917 * 10^{-3} kg, so about 8 gram which is very light.

The six holes in the stainless steel sheet have a total area $A_h = 6 * \pi/4 * 12^2 = 679$ mm^2. The water has to pass these holes but it also has to pass the circular gap in between the valve and the seat. This circular gap has a diameter of 44 mm at the outside of the 12 mm holes. So the gap area $A_g = \pi * 44 * 3 = 415$ mm^2. So $A_g < A_h$ and therefore A_g is critical and the highest water speed will be realised in the gap.

The flow which is pumped by one diaphragm is maximal half way the stroke. The crank radius r is about perpendicular to the axis of the connecting rod for this position. The diaphragm velocity V_d for this position is given by:

$$V_d = \omega * r = \pi * n * r / 30 \quad \text{(m/s)} \qquad (20)$$

As water is incompressible, the product of the diaphragm velocity V_d times the average diaphragm area A_{av} must be equal to the product of the water speed in the gap V_g times the gap area A_g. This results in:

$$V_g = V_d * A_{av} / A_g \quad \text{(m/s)} \qquad (21)$$

(9) + (20) + (21) gives:

$$V_g = \pi^2 * n * r * D_{av}^2 / (120 * A_g) \quad \text{(m/s)} \qquad (22)$$

Substitution of n = 277 rpm, r = e = 0.007 m, D_{av} = 0.06 m and A_g = 415 mm^2 = 4.15 * 10^{-4} m^2 in formula 21 gives V_g = 1.38 m/s. This is a rather high speed but it is assumed that this speed is allowed.

The calculated speed is the maximum speed half way the stroke of the diaphragm. The speed varies sinusoidal during the movement of the diaphragm. How the valve moves as the result of this varying water speed and the influence of the valve mass and the spring, is difficult to determine.

To get a rough impression, it is assumed that the movement of the valve follows the sinusoidal fluctuation of the water speed in the gap. This means that the water speed in the gap is constant if the water speed varies sinusoidal. The gap of a suction valve is becoming wider during the first halve upwards stroke, so during a quart revolution. The gap is becoming narrower during the second halve upwards stroke. The same counts for a pressure valve during the downwards stroke.

The sinusoidal movement of the valve is only valid for the time that water is flowing through the valve, so during a halve revolution of the crank shaft. The valve is closed during the other halve revolution. The valve stroke is 3 mm. The displacement of the connecting rod during a quart revolution is 7 mm. This means that the acceleration of the valve is a factor 3 / 7 = 0.429 times the acceleration of the diaphragm. The maximum acceleration of diaphragm was calculated to be 5.9 m/s^2. So the maximum acceleration of the valve is 0.429 * 5.9 = 2.5 m/s^2.

The real movement of the valve will be not sinusoidal. It can be expected that the valve will open in a short time, then rest against the stop for a certain time and then again will close in a short time. This will result in much higher acceleration forces than for a sinusoidal movement. Assume the real acceleration is a factor 5 higher so 12.5 m/s^2. Substitution of m = 0.0079 kg and a = 12.5 m/s^2 in formula 19 gives that F = 0.098 N which is a low force.

Opening of the valve will be caused by the water flow. The water flow will cause a certain pressure difference over the valve and an extra spring force is required to neutralise this pressure drop during closing. The weight of the valve has a tendency to close the suction valve and to open the pressure valve (see figure 2). However, a valve mass of 0.0079 kg results in weight of only 0.077 N which might be small with respect to the required force to neutralise the pressure difference over the valve. So the weight will have a limited influence. But it means that the spring has to be made as strong as possible for the available space.

An aspect which may be active is that the pressure valve will close after the bottom dead centre because of the inertia of the water. This means that there might be a flow through the pressure valves during the first part of the upwards stroke of the connecting rod. The same effect is active for the suction valve. This effect will result in a larger flow than the flow calculated with formula 5. The required power will also be larger and therefore it is nice that the chosen motor has a certain reserve.

The 1” hose pillars to which the suction and the pressure hoses are connected have an inside diameter of 25 mm. This results in a pillar area $A_p = \pi/4 * 25^2 = 491$ mm^2. Taking $V_g = V_p$ and $A_g = A_p$ makes that formula 22 changes into:

$$V_p = \pi^2 * n * r * D_{av}^2 / (120 * A_p) \qquad (m/s) \qquad (23)$$

Substitution of n = 277 rpm, r = e = 0.007 m, D_{av} = 0.06 m and A_p = 491 mm^2 = $4.91 * 10^{-4}$ m^2 in formula 23 gives V_p = 1.17 m/s. This seems an acceptable speed. It is assumed that the inside diameter of the suction and the pressure hoses are the same as the inside diameter of the 1” hose pillar, so also 25 mm. So the maximum water speed in the suction hose is about 1.17 m/s. However, because of the flexibility of the hose, the fluctuation of the water speed will be flattened as the distance in between the pump is larger. This will result in a little lower average speed for a diaphragm pump with three diaphragms. The flattening of the speed fluctuation is especially important for the pressure pipe because there is a large damping water mass in the long pipe which is used for the remaining plastic pressure pipe after the 4 meter flexible hose. The fluctuation in the 4 meter suction hose will be flattened only a little as there is no large water mass in it.

To know if the given pump will work nicely, a prototype has to be built and tested. For testing one needs a motor for which the rotational speed can be varied which makes it possible to start with a low rotational speed. One can use a frequency modulator if the original 4-pole, 3-phase motor is used. One also can use the prescribed permanent magnet DC motor but one has to limit the voltage up to maximal 22.4 V if a gear box with i = 5.06 is used! If the permanent magnet DC motor is the final motor, one has to order a gear box with i = 6.33 and in this case the maximum voltage can be 28 V.

The gear motors mentioned in the Rossi catalogue already contain an asynchronous motor. But it is possible to order a gear box without a motor but including the gear wheel which has to be mounted to the motor shaft.

7 References

1 Kragten A. Water pumping with a windmill, March 2012, modified September 2013, free public report KD 490, engineering office Kragten Design, Populierenlaan 51, 5492 SG Sint-Oedenrode, The Netherlands.

2 Kragten A. Membraanpomp Vahita 62 (in Dutch), October 1993, report KD 03, engineering office Kragten Design, Populierenlaan 51, 5492 SG Sint-Oedenrode, The Netherlands.

3 Kragten A. Ideas about a direct drive 22-pole permanent magnet generator for the VIRYA-3B2 windmill using the stator stamping of an Indian 4-pole, 3-phase, 3 kW asynchronous motor frame size 100 and 33 neodymium magnets size 40 * 10 * 5 mm, October 2012, report KD 506, engineering office Kragten Design, Populierenlaan 51, 5492 SG Sint-Oedenrode, The Netherlands.

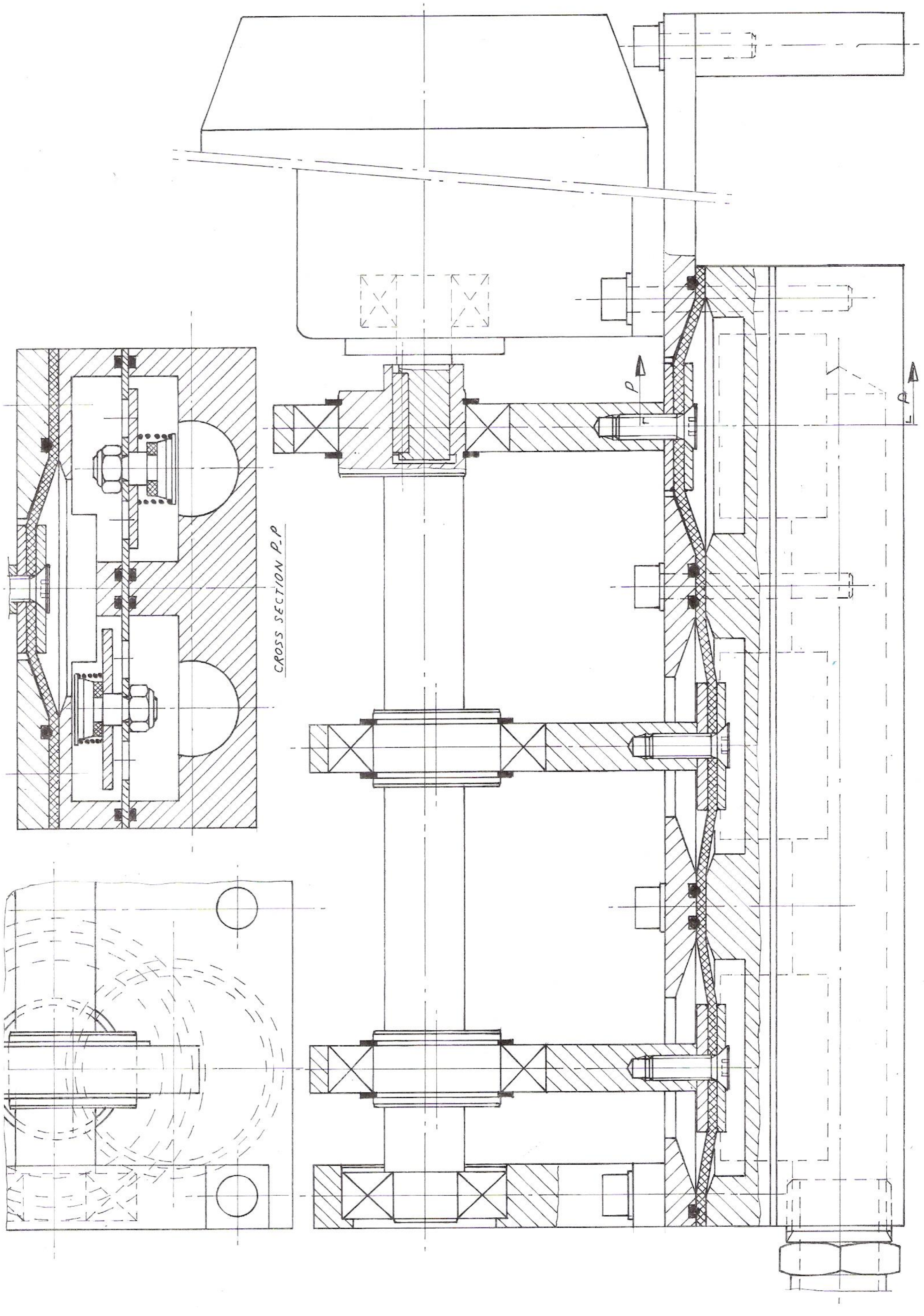

fig. 2 Diaphragm pump with three connecting rods