Ideas about a vane pump for low height irrigation directly driven by a 24 V, 0.35 kW, permanent magnet DC motor frame size 71

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KD 661

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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 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 the lowest pump price. The pump described in this report can be used for irrigation for a maximum static height H_{stat} = 10 m.

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 river or lake 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. A motor efficiency η_m of about 0.8 is possible for a small PM-motor. It is chosen to use a 24 V DC permanent magnet (PM) motor as 24 V is a common voltage and the cable losses are much smaller than for a 12 V motor. It is assumed that the motor is placed close to the energy source so the cable losses are limited. So it is assumed that the cable efficiency η_c = 0.97. An advantage of a DC motor is that it can directly be driven by a solar panel or a wind turbine.

It might be possible to use the pump without batteries if there is a voltage controller and dump load which limits the maximum voltage. In this case there must also be a relay in between the windmill generator and the pump motor which breaks the connection below a certain voltage to facilitate starting of the windmill. However, use of a battery makes that the generated energy can be used for other goals than water pumping if the reservoir is full.

Certain permanent magnet DC motors are supplied with the same frame size as used for asynchronous motors. Such a motor is chosen. The motor is described in chapter 5.

The highest pump efficiency η_p is possible for a positive displacement pump. An efficiency of maximum 0.9 has been measured for single acting piston pumps. But it is rather difficult to make a fast running piston pump with a non fluctuating flow. Therefore a rotating vane pump is chosen. The efficiency of a vane pump will be somewhat lower than for a piston pump because of internal water leakage. It is expected that a well designed vane pump will have an energetic pump efficiency η_p = 0.85 and a volumetric efficiency η_{vol} = 0.95 for the nominal rotational speed. The highest transmission efficiency η_{tr} is realised if there is no reducing transmission. So the vane pump is mounted directly to the motor shaft and η_t = 1.

So the maximum total efficiency η_{tot} of the pump and a DC, PM-motor including cables $\eta_{tot} = \eta_c * \eta_m * \eta_t * \eta_p = 0.97 * 0.8 * 1 * 0.85 = 0.66$ which is rather high. 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 height which depends on the flow. The pipe losses aren't incorporated in the efficiency of the pump-motor combination. It is assumed that the dynamic height is 10 % of the static height.

In 2013, I did some research to a fast running vane pump which is directly driven by an asynchronous motor. This pump is described in my public report KD 539 (ref. 1). This pump was designed for a static height of 40 m and it therefore has a rather small stroke volume. For irrigation it is assumed that the maximum static height is 10 m and then a much bigger stroke volume is needed if a motor with about the same power is used.

The vane pump geometry, the determination of the shape of the chamber in the housing and the determination of the stroke volume are described in chapter 2, 3 and 4 of KD 539. These three chapters are about copied in this new report KD 661 and therefore it isn't necessary to study KD 539. Several options how water can be pumped with a windmill are described in my public report KD 490 (ref. 2).

2 Description of the vane pump

The vane pump has a rotor in which two sleeves are made under an angle of 90°. Both sleeves have the same depth. One strip is positioned in two opposite sleeve ends. The strips jut a little out of the rotor. Every strip has a cut-out in the centre to prevent that the strips touch each other in the area in which they normally move. The strips are rounded at the outside with a radius which is smaller than the smallest radius of the chamber made in the pump housing. The rotor is turning in a housing which has a chamber which is eccentric with respect to the rotor axis. The shape of the chamber is such that the strips have a minimal clearance in the housing. The depth of the chamber is identical to the height of the vanes. The housing has a cover at the left side. The rotor has two bearings at the right part of the housing which makes mounting of the strips easy. The cover is provided by two kidney shaped openings through which the water is sucked and pressed. As there must always be one vane in between the suction and the pressure side of the pump, each kidney shaped opening covers an area of 90°. A sketch of such a vane pump is given in figure 1.

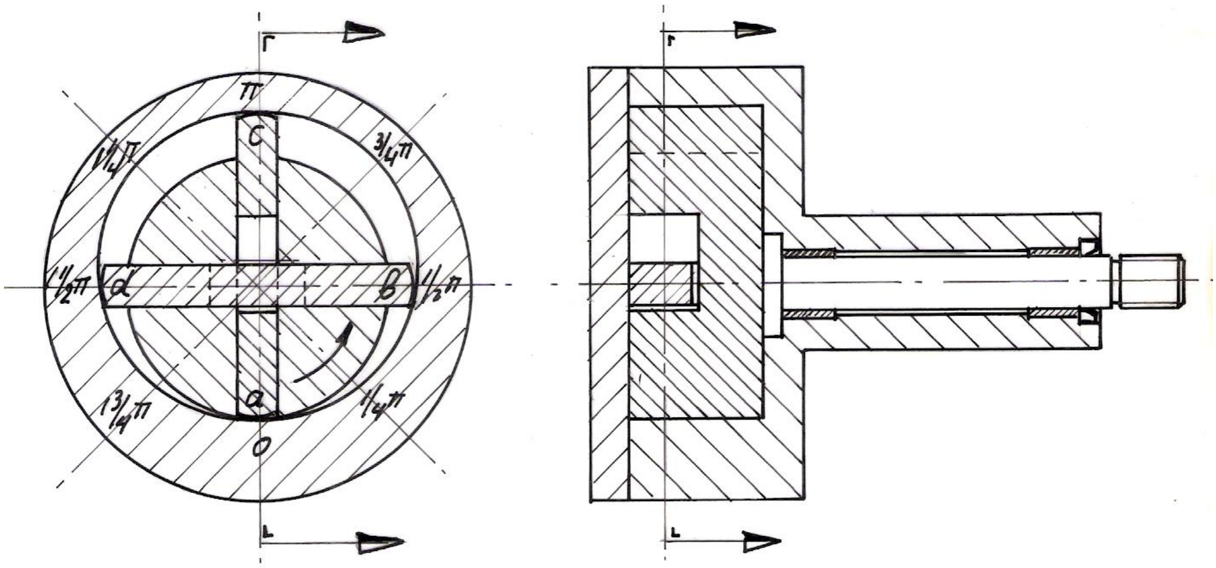

fig. 1 Sketch vane pump

The cut-out in each strip will be filled with water. This water has to be pressed from one side of the cut-out to the other side when the strip is moving but this isn't possible for the given construction as the flow is blocked by the other strip. This problem can be solved by making the cut-out a little deeper than half the vane height. Another option might be to make a shallow circular chamber at both sides of the rotor but this weakens the connection of the shaft to the rotor.

3 Determination of the shape of the chamber in the housing

In the left cross section of figure 1, one can distinguish two sectors which determine the functioning of the pump and the movement of the vanes. Each sector covers an area $\Delta\varphi = 180°$ or π radial. The right sector runs from 0 up to π rad. The left sector runs from π up to 2 π rad. The two strips are forming four vanes which are numbered a, b, c and d. Vane a and c are a part of one strip. Vane b and d are a part of the other strip. The rotor is turning left hand if seen from the left side. The rotor is drawn in figure 1 for a position where vane a coincides with $\varphi = 0$.

The right sector, which runs from $\varphi = 0$ up to $\varphi = \pi$, is the area in which vane a is moving to the outside. The space in between the rotor and the chamber is increasing and therefore water is sucked in this area. The kidney shaped suction opening is lying in this sector and covers the area from ¼ π up to ¾ π. The left sector, which runs from $\varphi = \pi$ up to $\varphi = 2\pi$, is the area in which vane c is moving to the inside. The space in between the rotor and the chamber is decreasing and therefore water is pressed in this area. The kidney shaped pressure opening is lying in this sector and covers the area from 1 ¼ π up to 1 ¾ π. Because both the suction and the pressure opening cover an area of ½ π, in between these areas there are also two areas which cover ½ π. Therefore there will be always at least one vane in between the suction and the pressure opening.

A prerequisite for the shape of the chamber in the housing is that the strips have minimal clearance for every position of the rotor. If this is realised, there will be minimal internal leakage of water, even at low rotational speeds. The chamber can't have the shape of a cylinder because for a cylinder, the available space for the position of vanes b and d is smaller than for the position of vanes a and c. For the determination of the shape, the chamber is seen as an internal cam. The radius at the end of each vane is replaced by a cam roller with the same radius. The angle over which the cam roller is moving is taken in radials. The shape of the left and the right part of the chamber must be such that the heart of the cam roller moves with the same distance to the outside in the right sector as it moves to the inside in the left sector. This is the case if for both sectors the same mathematical function is used which is 180° rotary symmetrical around the middle of the function.

In principle four functions fulfil this need being: a cosine, an inclined sine, a fifth degree polynomial and a seventh degree polynomial. For all four functions the radial speed is zero for $\varphi = 0$ and for $\varphi = \pi$. The cosine has the maximum acceleration for the beginning and the end of the function. But as both functions for the right and the left sector are symmetrical around the line through $\varphi = 0$ and $\varphi = \pi$, the same cosine function is needed for the left and for the right sector. Therefore there will be no jump in the acceleration at $\varphi = 0$ and $\varphi = \pi$. The cosine function has the lowest maximal acceleration and the smallest pressure angle in between the cam roller and the curve in the chamber. It also has the simplest mathematical description. The cosine function has therefore been chosen.

The diagram for the movement s, the speed v and the acceleration a as a function of φ is given in figure 2. Now the question is, what mathematical function has to be used for the curve of the left and the right section. It concerns the displacement function of the heart of the cam roller. During milling of the chamber, one has to use a cutter with the same diameter as the cam roller, so with the same radius as the radius at the end of the strip! The real shape of the curve which is followed by the heart of the cam roller is the displacement function of the heart of the roll unrolled on the so called basic circle.

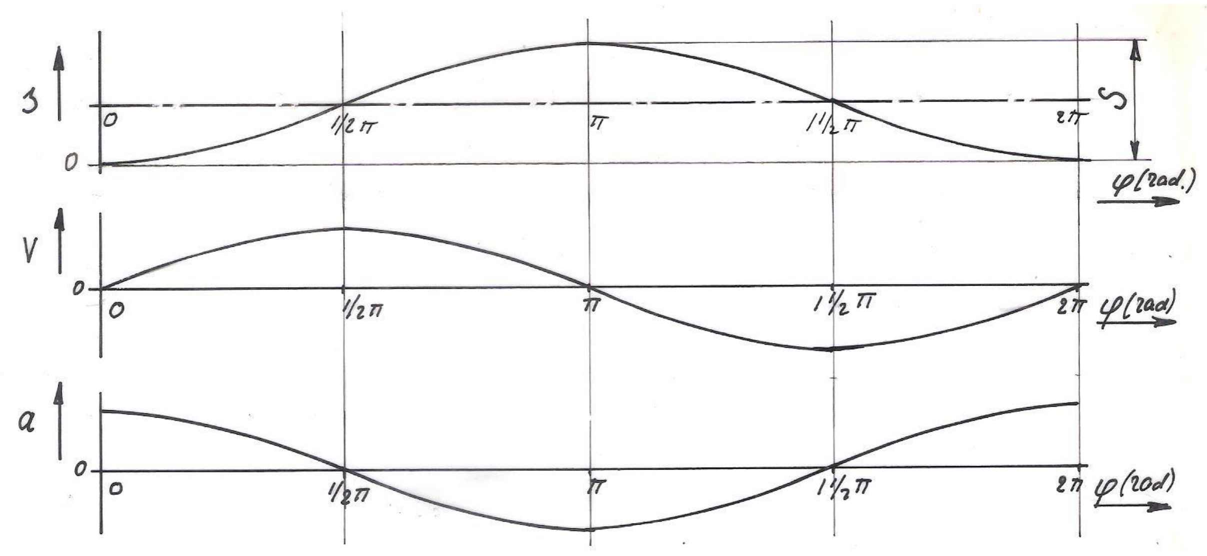

fig. 2 Variation s-φ, v-φ en a-φ curves for the centre of the roll which replaces the radius at the end of each vane.

The maximum stroke of a strip is chosen to be S. So the maximum stroke is the width of the chamber for $\varphi = \pi$. The displacement of vane a is called s and s = 0 for $\varphi = 0$. As the speed must vary according to a sine function and as the speed is the first derivative of the displacement, the displacement must be a $-\cos\varphi$ function. After some try and error it is found that the displacement s as a function of φ is given by:

$$s = f(\varphi) = \tfrac{1}{2} S - \tfrac{1}{2} S \cos\varphi = \tfrac{1}{2} S (1 - \cos\varphi) \qquad (1)$$

This gives as the first and the second derivative that:

$$f'(\varphi) = \tfrac{1}{2} S \sin\varphi \qquad (2)$$

$$f''(\varphi) = \tfrac{1}{2} S \cos\varphi \qquad (3)$$

$f'(\varphi)$ is representative for the variation of the speed v and $f''(\varphi)$ is representative for the variation of the acceleration a. However, to find the absolute values of v and a one has to take the derivative to the time t in stead of the angle φ. But for these derivatives, the rotational speed has to be taken into account. This isn't useful to determine the shape of the curve of the chamber.

Formula 1 is now checked for the beginning, the middle and the end of the curves of the right and the left section. $\varphi = 0$ rad gives s = 0. This is right. $\varphi = \frac{1}{2}\pi$ rad gives s = ½ S. This is right because here we are half way the right curve. $\varphi = \pi$ rad gives s = S. This is right. $\varphi = 1\frac{1}{2}\pi$ rad gives s = ½ S. This is right because here we are half way the left curve. $\varphi = 2\pi$ rad gives s = 0. This is right because here we are at the beginning again.

One has to be alert that formula 1 describes the function of the heart of the cam roller or the cutter of the milling machine and so it is not the function of the surface of the chamber which is made in the housing. The programmable milling machine has to be programmed such that the heart of the cutter follows the correct path!

4 Determination of the stroke volume and the flow

The theoretical stroke volume of the pump $\nabla_{p\ th}$ is the volume of water pumped during one revolution if there is no internal leakage. It is very tricky to determine the stroke volume exactly. This is because the stroke of the vanes varies continuously and because a small part of the pumped water is returned from the pressure side to the suction side. To be able to find a reasonable approximation, the rotor is drawn such that vane a coincides with $\varphi = ¼\ \pi$ (see figure 3). The suction and pressure openings are separated by two vanes for this position.

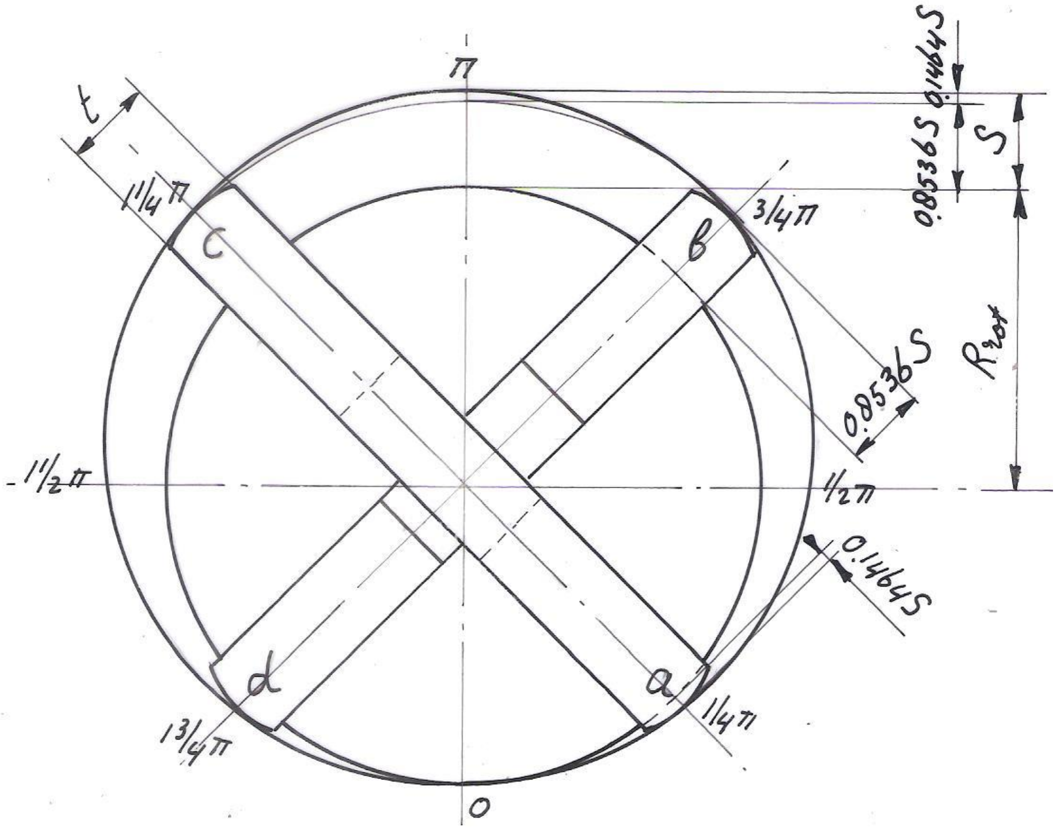

fig. 3 Rotor drawn for the position that vane a coincides with $\varphi = ¼\ \pi$

For a quart revolution, the volume of water which is contained in the space in between the rotor, the housing and the vanes b and c is pumped from the suction side to the pressure side. For a quart revolution, the volume of water which is contained in the space in between the rotor, the housing and the vanes d and a is pumped back from the pressure side to the suction side. This is only possible if there is a small gap in between the rotor and the chamber surface. So for a quart revolution the difference in between both volumes is pumped from the suction to the pressure side and four times this difference is pump during one complete revolution.

To be able to determine the volumes in between the vanes b and c and in between the vanes d and a, the areas in the plane of rotation in between the vanes, the rotor and the housing have to be determined. The problem for this determination is that s varies. To find a simple formula for $\nabla_{p\ th}$, an arc is drawn in the upper section with the hart of the rotor as centre and with such radius that the arc just touches the points where the vanes b and c are touching the chamber (see figure 3). For the right point at $\varphi = ¾\ \pi$, using formula 1, it can be determined that $s = ½\ S\ (1- \cos 3/4\pi) = 0.8536\ S$. The distance in between the arc and the curve in the housing for $\varphi = \pi$ then is S - 0.8536 S = 0.1464 S (see figure 3).

The area in between the vanes b and c is now divided into a part with a constant width of 0.8536 S and a moon shaped part with a maximum width of 0.1464 S.

The area in between the vanes d and a is formed by two half moon shaped areas which also have a maximum width of 0.1464 S. The summed length of both parts is somewhat shorter than the length of the moon shaped part in between the vanes b and c because the radius is smaller and because the vane is at the widest part. Therefore the area is also somewhat smaller. But if this difference is neglected, the amount of water pumped from the pressure side to the suction side is proportional to the moon shaped area in between the vanes b and c. So the net amount of water pumped in a quart revolution from the suction side to the pressure side is proportional to the area in between the vanes b and c with a width 0.8536 S.

The thickness of a vane is chosen t. The height of a vane is identical to the depth of the chamber in the housing and this height is chosen h. A vane has a tip radius R_{vane} which is identical to the radius of the cam roller for which the chamber is designed. R_{vane} must be smaller than the smallest radius of the chamber to prevent undercut. Therefore, R_{vane} is certainly smaller than the radius of the chamber at the positions $\varphi = ¼\ \pi$ and $\varphi = ¾\ \pi$. Therefore there will be two very small volumes at both sides of the contact line in between a vane and the chamber which also are a part of the pumped volume. These volumes are neglected and so it is assumed that a vane makes contact with the chamber over the whole thickness of the vane. The area of a vane which is jutting out of the rotor at $\varphi = ¾\ \pi$ then is about 0.8536 * t * S. In the following formulas all measures are taken in m. For the theoretical stroke volume $\nabla_{p\ th}$ we now find that:

$$\nabla_{p\ th} = \{\pi\ [\ (R_{rot} + 0.8536\ S)^2 - R_{rot}^2] - 4 * 0.8536 * t * S\} * h \qquad \text{or}$$

$$\nabla_{p\ th} = [\pi\ (1.7072\ R_{rot} * S + 0.7286\ S^2) - 3.4144 * t * S] * h \qquad (m^3) \qquad (4)$$

Because of the volumetric efficiency η_{vol}, the real stroke volume ∇_p is smaller than $\nabla_{p\ th}$. The volumetric efficiency is determined by the gaps in between the rotor, the covers, and the vanes and by the height and the rotational speed. For certain gaps, the leaking flow is almost independent of the rotational speed. The volumetric efficiency will therefore become lower as the rotational speed drops. It is assumed that the pump can be designed such that the volumetric efficiency is about 0.95 for the design rotational speed and the design height. The real stroke volume ∇_p is given by:

$$\nabla_p = \nabla_{p\ th} * \eta_{vol} \qquad (m^3) \qquad (5)$$

The flow q is the real stroke volume times the number of revolutions per second of the pump. The number of revolutions of the pump n is normally given in rpm. This gives:

$$q = \nabla_p * n / 60 \qquad (m^3/s) \qquad (6)$$

(4) + (5) + (6) gives:

$$q = [\pi\ (1.7072\ R_{rot} * S + 0.7286\ S^2) - 3.4144 * t * S] * h * \eta_{vol} * n / 60 \qquad (m^3/s) \qquad (7)$$

5 Description of the PM-motor

There are manufactures of permanent magnet DC generators which make use of the same frame size as used for asynchronous motors. One of these manufacturers is the Dutch company Creusen from Roermond (see www.creusen.nl). Permanent magnet motors of this brand are supplied for frame size 56, 71 and 90. Assume frame size 71 is chosen. A range of voltages and powers is available for frame size 71. Assume the nominal voltage is 24 V and the nominal power is 350 W at a rotational speed of 1500 rpm. This requires motor housing 71L-2GP 24 V. If this motor is used in combination with a solar panel or a windmill, the real loaded voltage will be higher than 24 V. It is assumed that a voltage controller is used which limits the voltage up to 28 V. If the current is constant, it means that the power is increased by a factor 28 / 24 and becomes about 408 W. This also means that the maximum rotational speed is about 1500 * 28 / 24 = 1750 rpm. This is a factor 1.25 higher than the loaded rotational speed of a 4-pole asynchronous motor for which the nominal rotational speed is about 1400 rpm. If a grid is available, the PM-motor can be replaced by an asynchronous motor but the further calculations are given for a PM-motor running at 1500 rpm.

Motors of frame size 71 are normally supplied with a foot B3 but can also be supplied by a flange. The flange version is chosen and the flange has size C105. The pump is directly connected to the flange. 105 is the outside diameter of the flange in mm. It is provided by four threaded holes M6 at a pitch circle of 85 mm. It has a centralization collar with a diameter of 70 mm and a height of the collar of 2.5 mm.

A motor with frame size 71 has a shaft diameter d = 14 mm. The shaft diameter at the bearings is 15 mm so there is a 0.5 mm collar. The shaft length is 30 mm. There is a 5 mm key groove in the shaft. The key juts 2 mm out of the shaft.

The pump will be designed such that it has a hollow shaft with a key groove and that the motor axis can simply be pushed in the hollow shaft end of the pump. The pump must have a flange at the back side which is bolted to the flange of the motor by four M6 bolts.

The pump will be mounted with the pump shaft vertical and the motor on top of it. This has as advantage that there is no risk that the motor and the pump bearings become wet because of water leaking along the pump shaft. The inlet and the outlet opening of the pump are horizontal and opposed to each other. The pump must have a foot at the bottom side for placing the pump on the ground.

6 Determination of the pump geometry and the output

It is assumed that the motor isn't used at the nominal power of 350 W at 1500 rpm but at a power of 320 W at 1500 rpm. So there is some reserve if the pump efficiency is lower than expected. The required electrical power depends on the motor efficiency and the cable losses. It is assumed that the motor efficiency is 0.8 and that the cable efficiency is 0.97. So the required electrical power is 320 / (0.8 * 0.97) = 412 W. The hydraulic power P_{hyd} is given by:

$$P_{hyd} = \rho_w * g * H * q \qquad \text{(W)} \qquad (8)$$

In this formula H is the total height. H is the sum of the static height H_{stat} and the dynamic height H_{dyn}. The static height is the height in between the water level in the river or lake 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 height depends on the flow, the length of the pipes and very much on the inside pipe diameter. Calculation of the dynamic height is out of the scope of this report. It is assumed that the dynamic height is 10 % of the static height.

The required mechanical power at the motor shaft P depends on the pump efficiency η_p and is given by:

$P = P_{hyd} / \eta_p$ (W) (9)

(8) + (9) gives:

$P = \rho_w * g * H * q / \eta_p$ (W) (10)

(5) + (6) gives:

$q = \nabla_{p\ th} * \eta_{vol} * n / 60$ (m^3/s) (11)

(10) + (11) gives:

$P = \rho_w * g * H * \nabla_{p\ th} * \eta_{vol} * n / (60 * \eta_p)$ (W) (12)

Formula 12 can be written as:

$\nabla_{p\ th} = 60 * P * \eta_p / (\rho_w * g * H * \eta_{vol} * n)$ (m^3) (13)

It is assumed that the pump is designed for a static height H_{stat} = 10 m. It is assumed that the dynamic height H_{dyn} = 1 m. This gives a total height H = 11 m. Substitution of P = 320 W, η_p = 0.85, ρ_w = 1000 kg/m^3, g = 9.81 m/s^2, H = 11 m, η_{vol} = 0.95 and n = 1500 rpm in formula 13 gives that $\nabla_{p\ th}$ = 106.1 * 10^{-6} m^3 = 106.1 * 10^3 mm^3 = 0.1061 litre. This is about a factor 3.72 larger than the stroke volume of the pump which is described in report KD 539 for which ∇_{pth} = 28.5 * 10^{-6} m^3. This means that all pump dimensions of this pump have to be scaled up by about a factor 3.72 $^{1/3}$ = 1.55 to get the correct stroke volume.

In the first instance it is assumed that the scale factor is 1.5. The main dimensions of the pump described in KD 539 are: R_{rot} = 25 mm, S = 9 mm, t = 8 mm, R_{vane} = 10 mm and h = 25 mm. This results in a rotor radius R_{rot} = 1.5 * 25 = 37.5 mm, in a stroke S = 1.5 * 9 = 13.5 mm, in a vane thickness t = 1.5 * 8 = 12 mm, in a vane radius R_{vane} = 15 mm and in a vane height h = 1.5 * 25 = 37.5 mm. So for this scale factor, the stroke volume increases by a factor 1.5^3 = 3.375 which is a bit too low. Assume that S is chosen 14 mm and that h is chosen 40 mm. So now the stroke volume increases by a factor 3.375 * 14/13.5 * 40/37.5 = 3.73 which is about right. Up to now it is chosen that R_{vane} = 15 mm and this requires a rather large 30 mm cutter. R_{vane} can be chosen larger and this will reduce the wear of the vane tip but larger cutters may not be available. The basic circle will be smaller for a larger value of R_{vane}.

Formula 4 can be used to check if the theoretical stroke volume is correct for the chosen dimensions. Substitution of R_{rot} = 37.5 mm, h = 40 mm, t = 12 mm and S = 14 mm in formula 4 gives $\nabla_{p\ th}$ = 107.6 * 10^3 mm^3 = 107.6 * 10^{-6} m^3. This is only a little larger than the calculated value of 106.1 * 10^{-6} m^3 so the chosen values for R_{rot}, h, t and S are correct.

The flow q at n = 1500 rpm can now also be determined. Substitution of $\nabla_{p\ th}$ = 107.6 * 10^{-6} m^3, η_{vol} = 0.95 and n = 1500 rpm in formula 11 gives q = 2.555 * 10^{-3} m^3/s = 153.3 litre/min = 9.2 m^3/hour = 220.8 m^3/day. This is a large amount of water and this requires a large reservoir if the needed power is constantly available. This is certainly not the case if the power is supplied by solar panels but the calculation shows that the flow is large and so the pump can be used for irrigation. If the pump motor is running at 1750 rpm in stead of 1500 rpm, the flow is about a factor 7/6 higher so about 10.7 m^3/hour. If the static height H_{stat} is less than 7 m, the required power is at least a factor 0.7 lower than for H_{stat} = 10 m. So a smaller and cheaper 250 W pump motor can be used. The type number of this motor of manufacture Creusen is 71M-2GP 24 V.

In the calculations up to now it is assumed that there is no gap in between the rotor and the chamber at $\varphi = 0$. In reality this isn't possible because the water of the left half moon shaped area in between vane d and vane a is pumped from the pressure side to the suction side. So there must be a little gap in between the rotor and the housing at $\varphi = 0$ otherwise an infinite pressure will be built up. So the real value of the rotor must be a little smaller than 37.5 mm. If the real value is chosen 37.2 mm, the rotor diameter will be 74.4 mm and this means that the rotor can be machined from 75 mm stainless steel bar. But for the calculations, the value $R_{rot} = 37.5$ mm is used.

To make the shape of the chamber one needs a programmable milling machine. One has to use a finger cutter with a radius $R = R_{vane} = 15$ mm, so with a diameter of 30 mm. This is a large diameter for a chamber depth of 40 mm, so milling of the chamber can be done very stable. The stroke $s = 0$ for $\varphi = 0$. The shape of the curve which has to be followed by the heart of the cutter is the displacement function of the roller unrolled on the basic circle. This means that the diameter of the basic circle is 75 – 30 = 45 mm for $R_{vane} = 15$ mm and for $R_{rot} = 37.5$ mm. Formula 1 gives the stroke as a function of φ. Substitution of S = 14 mm in formula 1 gives that:

$$s = 7\,(1 - \cos\varphi) \quad \text{(mm)} \tag{14}$$

s is now calculated for values of φ starting at $\varphi = 0°$ and increasing by 15°. The result of the calculation is given in table 1. The radius of the heart of the cutter r_c is 22.5 mm larger than s if the basic circle has a diameter of 45 mm.

φ (°)	s (mm)	r_c (mm)
0	0	22.5
15	0.239	22.739
30	0.938	23.438
45	2.050	24.550
60	3.5	26
75	5.188	27.688
90	7	29.5
105	8.812	31.312
120	10.5	33
135	11.950	34.450
150	13.062	35.562
165	13.761	36.261
180	14	36.5
195	13.761	36.261
210	13.062	35.562
225	11.950	34.450
240	10.5	33
255	8.812	31.312
270	7	29.5
285	5.188	27.688
300	3.5	26
315	2.050	24.550
330	0.938	23.438
345	0.239	22.739
360	0	22.5

Table 1 calculated values of s and r_c as a function of φ

The position of the cutter is given in figure 4 for φ increasing by 15°. The final shape of the chamber is a curved line which touches all the 30 mm circles representing the cutter at the outside. The final shape looks very much the same like a circle but there is a slight difference. Circles were used to draw figure 1 and 3.

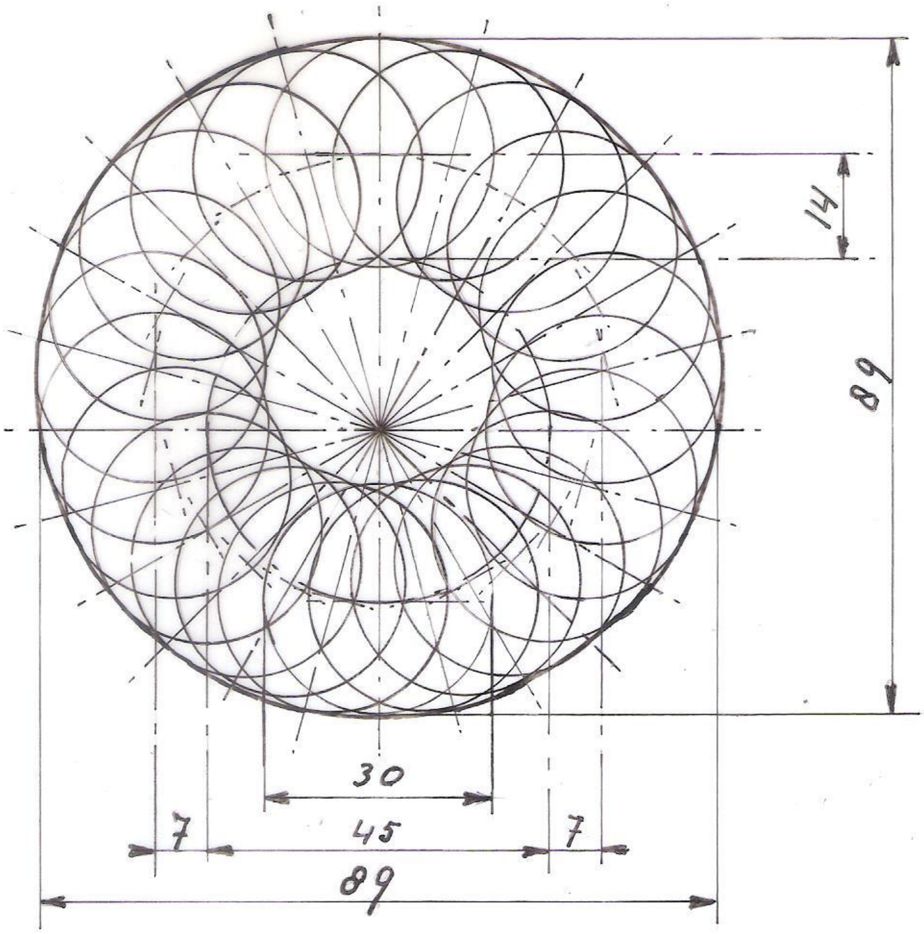

fig. 4 Positions of the cutter of a programmable milling machine for φ increasing by 15°

If a programmable milling machine isn't available, it might be possible to make the chamber shape using a turntable. If the pump housing is connected to the turntable with an eccentricity of exactly 7 mm and if a 30 mm cutter is used with the correct starting position, the required chamber shape can be realised by 360° rotation of the turntable and by a fixed position of the cutter.

The two grooves in the rotor have a width t = 12 mm and a height h = 40 mm. So the ratio h / t = 3.333 and this ratio is small enough for making an accurate groove with parallel sides. R_{rot} = 37.5 mm and S = 14 mm. So the length of vane is 2 * 37.5 + 14 = 89 mm. Each vane is provided by a cut-out in the middle to prevent that the strips make contact. The width of this cut-out is 28 mm. Assume that the depth of the cut-out is made 22 mm, so a little more than half the vane height to prevent blockage of the internal flow.

The strip volume is about 76 * 40 * 12 = 36480 mm^3. The strips are made out of a good quality plastic like POM or polyoxmethyleen (supplied as Delrin, Ertacetal or Hostaform). The density ρ is about $1.4 * 10^3$ kg/m^3 or $1.4 * 10^{-6}$ kg/mm^3. So the mass of a vane is about 0.051 kg = 51 gram which is very low. If POM is wearing too fast one may use carbon imbedded Teflon or may be even a ceramic material.

The housing is made out of stainless steel bar with a diameter of 140 mm. A 105 mm flange is made at the top side to be able to connect the pump to the motor flange. The rotor shaft must have a diameter of at least 25 mm if it is provided with a 14 mm hollow shaft and with a key groove for the motor shaft. So the geometry of the housing and the shaft differs from the sketch as given in figure 1.

The pump has no valves and small gaps in between the internal components. So the water will slowly flow down to the well once the motor is stopped. Another aspect is that the electric motor may turn in the opposite direction because the pump can work as a hydraulic motor driven by the pressure drop over the vanes. Both effects can be prevented if a foot valve is placed at the entrance of the suction line. But one must be alert that the motor never turns in the opposite direction if a foot valve is placed because this will blow the suction line or it will burn the motor winding!

A vane pump needs clean water with no sand particles. So a filter with a sufficient area to prevent a large pressure drop has to be placed at the entrance of the suction line. This filter can also contain the foot valve.

7 Calculation of the maximum acceleration force F_{max} acting on a strip

The second derivative f''(φ) given by formula 3 is representative for the acceleration. However, to find the real acceleration a, one has to take the derivative to t in stead of to φ. The relation in between φ, ω and t is given by:

$$\varphi = \omega * t \qquad (s) \qquad (15)$$

The relation in between the angular velocity ω and the rotational speed n in rpm is given by:

$$\omega = \pi * n / 30 \qquad (rad/s) \qquad (16)$$

(1) + (15) gives:

$$s = f\ (t) = ½\ S – ½\ S \cos(\omega * t) = ½\ S\ [1 – \cos(\omega * t)] \qquad (m) \qquad (17)$$

This gives as the first and the second derivative that:

$$v = f'\ (t) = ½\ S * \omega \sin(\omega * t) \qquad (m/s) \qquad (18)$$

$$a = f''\ (t) = ½\ S * \omega^2 \cos(\omega * t) \qquad (m/s^2) \qquad (19)$$

The acceleration a has an extreme value for φ = ω * t = 0 and for φ = ω * t = π (see figure 2). Both extremes have the same absolute value so any of the two values for φ can be taken. Substitution of ω * t = 0 in formula 19 gives for the maximum acceleration a_{max} that:

$$a_{max} = ½\ S * \omega^2 \qquad (m/s) \qquad (20)$$

The maximum acceleration force F_{max} is given by:

$$F_{max} = a_{max} * m \qquad (N) \qquad (21)$$

(20) + (21) gives:

$$F_{max} = ½\ S * \omega^2 * m \qquad (N) \qquad (22)$$

It was assumed that the maximal loaded rotational speed of the PM-motor n = 1750 rpm. Substitution of n = 1750 rpm in formula 16 gives that ω = 183.3 rad/s. Substitution of S = 14 mm = 0.014 m, ω = 183.3 rad/s and m = 0.051 kg in formula 22 gives F_{max} = 12 N. This is a low force, so moving of the vane will cause only little friction in between the vane and the chamber. The calculated force is equal to the imbalance force of one vane. The total imbalance force of two vanes will be a factor $\sqrt{2}$ larger, so about 17 N. As the pump has a rather large mass, it is expected that the vibrations caused by this imbalance force are acceptable.

8 Determination of the bearings, the seal and the hose pillars (see figure 5 page 17)

Most centrifugal pumps use a so called mechanical seal to prevent that the water is leaking out of the pump at the shaft. A mechanical seal is a rather complicated component for which the seal area is perpendicular to the rotor shaft. The mechanical seal is mounted in the pump housing as close as possible to the rotor and locked by a separate flange. Mostly ball bearings are used for the rotor shaft. The bearings are mounted in the dry part of the pump. For the vane pump, a different seal and different bearings are chosen.

It is chosen to use sealed needle bushes of manufacture INA with an inside diameter of 25 mm. The front bearing has a length of 30 mm. The bearing code is HK2530.2RS. The back bearing has a length of 20 mm. The bearing code is HK2520.2RS. The distance in between both bearings is chosen 30 mm. The distance in between the heart of the bearings is 55 mm. The bearings are used without inner rings so they are running directly on the stainless steel shaft. It is assumed that normal stainless steel is hard enough for the given load and rotational speed but it must be machined to a very flat surface with a very low roughness.

In stead of a mechanical seal, a Garlock PS-seal is chosen. This seal has the same dimension as a normal oil seal. It has a sealing lip which runs on the rotor shaft. The shaft diameter at the seal is 25 mm. The seal code is Garlock PS-seal 25 * 35 * 8. The sealing lip is made of a material called GYLON which is a modified PTFE. The shaft should be very smooth and rather hard. It is expected that normal stainless steel is hard enough for the chosen pump height and the chosen rotational speed of the pump. Information about this seal can be found on: www.eriks.nl/documentatie/afdichtingen/asafdichtingen/garlock-ps-seal.pdf. Tests have to prove if standard stainless steel is hard enough for the seal and for the bearings. If not, the shaft has to be covered by a hard layer which may be chromium or ceramics.

The inside diameter of the lip is much smaller than the shaft diameter when the seal isn't mounted. Therefore the shaft must have a tapered end when the shaft is mounted such that the shaft enters the seal from the pressure side. It is advised to use a cone angle of 15° and to make the smallest diameter of the cone 5 mm smaller than the diameter of the shaft. This results in a tapered length of about 9 mm.

The rotor shaft is mounted directly to the motor shaft. The motor shaft has a diameter of 14 mm and a length of 30 mm and it is provided with a 5 mm key which juts 2 mm out of the shaft. This key requires an internal key groove in the motor shaft. To be able to make this key groove a chamber with a diameter of 19 mm is required at the end of the hole. The key groove requires a shaft diameter of at least 25 mm at the bearings and that's why this diameter is chosen. This shaft has a 6 mm wide, 45° tapered cone at the back side of the rotor to make that the connection of the rotor and the shaft strong is enough.

The pump height results in a pressure difference Δp over the rotor. The pressure difference results in a force F which pushes the rotor to one side. This force is taken mainly by the front bearing. The pressure difference Δp is given by:

$$\Delta p = \rho_w * g * H \qquad (N/m^2) \tag{23}$$

Substitution of ρ_w = 1000 kg/m^3, g = 9.81 m/s^2 and H = 11 m in formula 23 gives Δp = 107910 N/m^2 = 0.10791 N/mm^2.

This pressure difference is working over the vane which has a length of 89 mm and a height of 40 mm. So the area of the vane A_v = 40 * 89 = 3560 mm^2.

The force F is given by:

$$F = \Delta p * A_v \quad \text{(N)} \tag{24}$$

Substitution of Δp = 0.10791 N/mm^2 and A_v = 3560 mm^2 in formula 24 gives F = 384 N. This force is working in the middle of the vane, so at a distance of 20 mm from the bottom of the chamber in the housing.

The distance f_1 in between the middle of the vane and the middle of the upper bearing f_1 = 110 mm. The distance f_2 in between the middle of the lower bearing and the middle of the upper bearing f_2 = 55 mm. The reaction force working on the lower bearing is called F_A. The reaction force working on the upper bearing is called F_B. Balance of moments around the heart of the upper bearing gives:

$$F_A = F * f_1 / f_2 \quad \text{(N)} \tag{25}$$

Substitution of F = 384 N, f_1 = 110 mm and f_2 = 55 mm in formula 25 gives F_A = 768 N.
$F_B = F_A – F$, so F_B = 384 N. So the load on the lower bearing is a factor 2 larger than the load on the upper bearing. The width of the lower bearing is chosen 30 mm and the width of the upper bearing is chosen 20 mm. The dynamic load factor C and the static load factor C_0 are given in the INA catalogue.
For bearing HK2530.2RS it is given that C = 25500 N and C_0 = 45000 N.
For bearing HK2520.2RS it is given that C = 15600 N and C_0 = 24000 N.
The ratio in between C is 1.635. The ratio in between C_0 is 1.875. Both ratios are smaller than 2 which indicates that the lower bearing is loaded heaviest. The calculated load on the lower bearing F_A = 768 N. This is very much lower than C_0, so the static load is absolutely no problem. The dynamic load factor C is used to calculate the lifetime of the bearing. The lifetime of a needle bearing in hours L_h is given by formula:

$$L_h = 16666 * (C / P)^p / n \quad \text{(hours)} \tag{26}$$

In this formula P is the load F_A. The coefficient p is 10/3 = 3.3333 for needle bearings. n is the rotational speed in rpm. Substitution of C = 25500 N, P = 768 N, p = 3.3333 and n = 1500 rpm in formula 26 gives L_h = 1307031 hours = 149.2 year. So the lower bearing is certainly strong enough. The lifetime of the upper bearing will be even longer. Concerning the required lifetime, lighter bearings could have been used but it is nice to have a large reserve because the bearings are used directly on a stainless steel shaft which isn’t hardened.

The Garlock PS-seal can have a nominal maximum pressure of 10 bar which corresponds to a height H = 100 m but the real maximum pressure depends on the circumference speed v of the shaft. This speed is given by:

$$v = d * \pi * n / (60 * 10^3) \quad \text{(m/s)} \tag{27}$$

Substitution of d = 25 mm and n = 1500 rpm in formula 27 gives v = 1.96 m/s. The allowable pressure depends on the type of GYLON used for the lip of the seal. Graphs for two types of GYLON are given being GYLON-B and GYLON-W.

GYLON-B is the standard material which is used for the chosen seal. In the graph it can be read that the allowable pressure for GYLON-B and v = 1.96 m/s is 10 bar. The maximum allowable speed for a pressure of 10 bar is even about 4 m/s so there is a large reserve. It is expected that a seal made out of GYLON-B will work if the roughness of the shaft is made low enough. After certain time of operation it has to be checked if the stainless steel is not worn at the position of the seal.

It has to be prevented that the lower bearing becomes wet if some water is leaking along the seal. Therefore two holes are drilled in the housing which ends in the space in between the seal and the lower bearing. An O-ring is used in between the housing and the front cover. The cover is connected to the housing by four inner hexagon bolts M8.

As needle bearings can’t take an axial load, the weight of the armature and the vanes is pushing on the cover. The armature is made of stainless steel but I expect problems if the cover is also made out of stainless steel. So the cover is made out of bronze. Cast bronze is available for a diameter of 142 mm and it isn’t necessary to machine the outer diameter.

The water can be transported through pipes or through hoses. Pipes will be used if a non flexible connection is wanted in between the pump and the reservoir. The advantage of using hoses is that they are flexible. Using a flexible suction hose makes it easy to prime the pump when it is running for the first time. Using a flexible pressure hose makes it possible to guide the hose directly to the place on the field where water is needed. So in this case one may work without a reservoir. Assume 1 ½” hoses are used. This means that the cover has to be provided with 1 ½” stainless steel hose pillars which are screwed horizontally in the left and the right side of the cover to make that the suction and the pressure hose are in parallel to the earth surface. So the cover must be rather thick.

1 ½” hose pillars have an inner diameter of 40 mm so an inner area $A_h = \pi/4 * 40^2 = 1257\ mm^2 = 1.275 * 10^{-3}\ m^2$. It is assumed that hoses with the same inner diameter are used. The water speed in the hose v_h is given by:

$$v_h = q / A_h \quad (m/s) \tag{28}$$

Substitution of $q = 2.555 * 10^{-3}\ m^3/s$ and $A_h = 1.275 * 10^{-3}\ m^2$ in formula 28 gives that $v_h = 2$ m/s. This seems an acceptable water speed for a flexible hose. The kidney shaped openings in the cover have to be made large and deep enough to prevent that high water speeds are created there. A cross section scale 1 : 1 of the pump is given in attachment 1. This picture is only meant to give an impression of the pump. Detailed drawings are needed for manufacture of a prototype.

As the pump is rather small it might be sensible to theft. It might be possible to provide the pump with couplings for which the hoses can be removed easily. In this way the pump can be taken home by night when nobody is working on the field.

9 References

1 Kragten A. Ideas about a fast running vane pump directly driven by a 0.37 kW, 4-pole asynchronous motor frame size 71, December 2013, reviewed September 2018, free public report KD 539, engineering office Kragten Design, Populierenlaan 51, 5492 SG Sint-Oedenrode, The Netherlands.

2 Kragten A. Water pumping with a windmill, March 2012, reviewed October 2018, free public report KD 490, engineering office Kragten Design, Populierenlaan 51, 5492 SG Sint-Oedenrode, The Netherlands.

10 Attachment 1

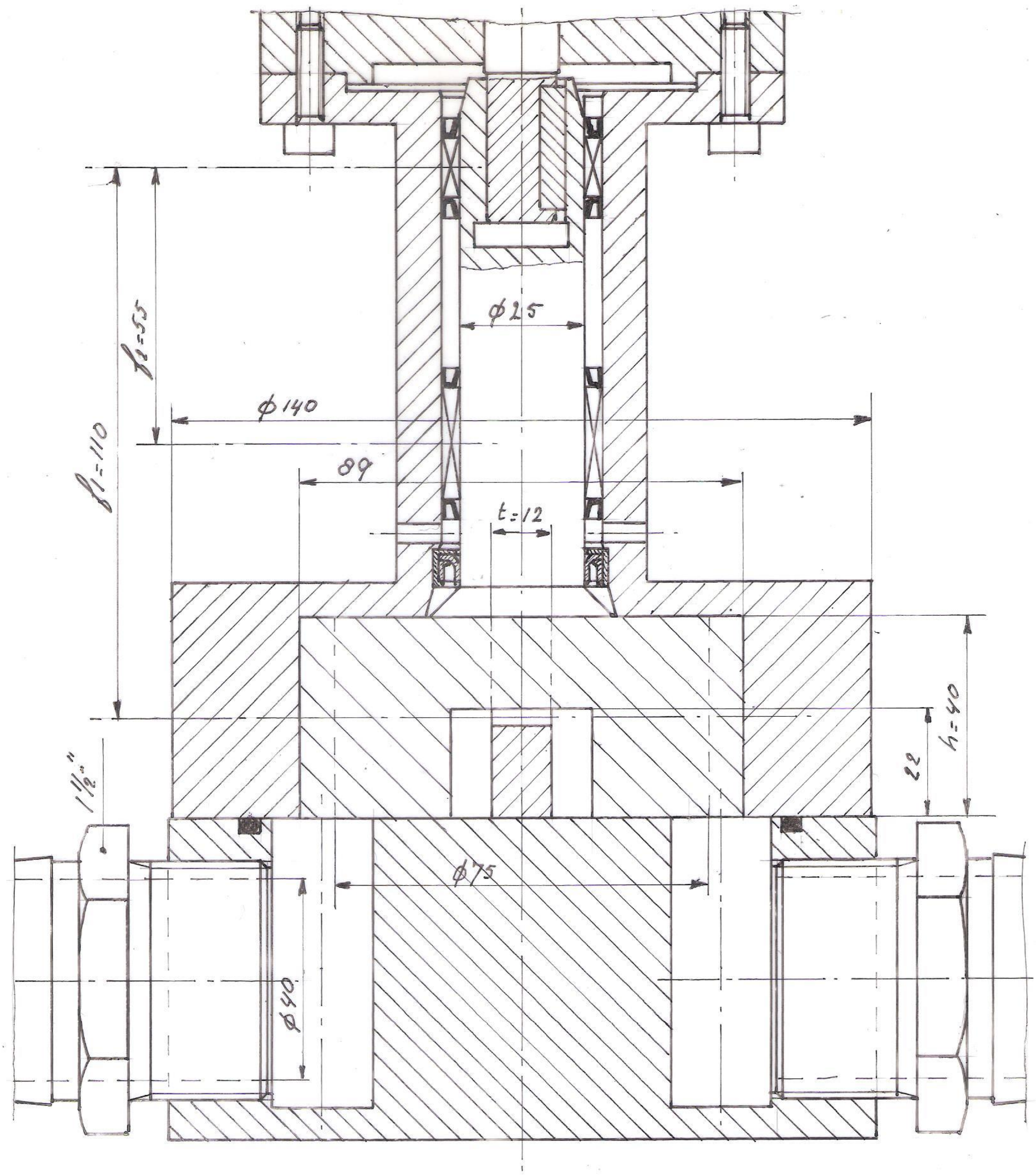

fig. 5 Cross section vane pump for connection to a DC PM-motor frame size 71