TECHNICAL

REPORT

1990

DIEVER 450

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FOREWORD

This technical report is the result of four years of research.
It is meant for people of the WOT to develop their knowledge
about waterpumping windmills (= windpumps) and especially
about the Diever 450 windmill. It is also meant for people in
developing countries who have the idea to built the Diever
450. To understand the theory and conceptions which are used
in the report the author assumes that the reader has
sufficient basic knowledge of wind energy. If this is not the
case the author advises to read first the WOT-publication
'wind energy for the third world' (lit.1).
The report reflects the development of the design and the
results of the tests on the prototype.

The author is grateful to the people who helped to design the
Diever 450, especially to A. de Roest, A. Schaap, C. Vos, S.
Vreeland, F. Van Oostrum, J. Andringa, G. Vlogman, B. de Jong,
G. Wijbenga and R. van leeuwen.

Frans Brughuis
may 1990

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LIST OF SYMBOLS

symbol	description	unit
A_{hs}	side area head	m²
A_p	piston area	m²
A_{pr}	area of the pumprod	m²
A_{pri}	area of the pumprod iron	m²
A_{proj}	projected area	m²
A_r	rotor area	m²
A_{rm}	area of the rising main	m²
A_{rs}	side area rotor (= 7,5% of A_r)	m²
B	number of blades	
c	corde	m
C_d	drag coefficient	
C_l	lift coefficient	
C_p	power coefficient	
C_Q	torque coefficient	
C_{Qstart}	starting torque coefficient	
D_r	rotor diameter	m
F	force	N
$\bar{F}$	admissible force	N
F_{peak}	peak force in the pumprod	N
F_{pr}	pumprod force	N
H	waterlifting head	m
l	blade length	m
l_k	buckling length	m
P	power	W
P_{out}	power output	W
P_r	mechanical power of the rotor	W
P_{wind}	power input	W
Q	torque	Nm
Q_{start}	starting torque	Nm
r	local radius	m
R_r	rotor radius	m
R_c	crank length	m
R_K, R_L,	reaction forces in	
R_M, R_N	K, L, M and N	N
s	stroke length	m

v	windspeed	m/s
v_d	design windspeed	m/s
v_{max}	maximum windspeed	m/s
v_r	rated windspeed	m/s
v_{start}	starting windspeed	m/s
v_{tip}	tipspeed	m/s

α	angle of attack	
β	blade angle	
γ	angle between actual position of main vane and its rest position	
δ	angle of yaw (between rotor axis and wind direction)	
ϵ	angle between hinge axis and vertical axis	
ϕ	angle between rotor plane and relative flow speed	
λ	tipspeed ratio	
λ_d	design tipspeed ratio	
λ_r	local speed ratio at radius r	
η_{hydr}	hydraulic efficiency of the pump installation	
η_{inst}	efficiency of the pump installation	
η_{mech}	mechanical efficiency of the pump	
η_{tot}	total efficiency of the windpump	
η_{tr}	efficiency of the transmission	
η_{vol}	volumetric efficiency of the pump	
Ω	rotor angular speed	rad/s
ρ_a	density of air	kg/m^3
ρ_c	density of concrete	kg/m^3
ρ_w	density of water	kg/m^3
σ	stress	N/mm^2
$\bar{\sigma}$	admissible stress	N/mm^2
σ_e	equivalent stress	N/mm^2
σ_{min}	minimum stress	N/mm^2
σ_{max}	maximum stress	N/mm^2
σ_v	surface stress	N/mm^2
τ	shear stress	N/mm^2
ω	buckling factor	

CHAPTER 1 INTRODUCTION

In 1979 the WOT designed the 12PU500 windmill. This windmill
is built in India, Indonesia, Bolivia, China, Tanzania and
other countries.
After seven years of experiences, in 1986, it was clear that
the 12PU500 design could be improved to get more output and a
longer life-time.
Therefore the WOT decided to redesign the windmill which
resulted in a new windmill, the 18PU450, also called the
Diever 450. Diever is the surname of our oldest and most loyal
member.

1.1 The 12PU500 windmill

The 12PU500 windmill is a
waterpumping windmill. It has
an horizontal-axis rotor with
12 blades and a diameter of
five meters. The tipspeed-
ratio λ (= tipspeed of the
rotorblades divided by the
windspeed, see lit. 1) is
two. The tower is a welded
construction of angle-irons
and at the top a pipe of 4",
the towerpipe. The tower has
a height of six meters. The
tail of the head construction
carries a windvane which
turns the head around the
towerpipe in order to keep
the rotor perpendicular to
the wind. The safety system
is half-automatic. In case of
a severe storm it will unlock

figure 1.1: the 12PU500
windmill

automatically the head from the tail after which the head and
rotor turns out of the wind. After the storm the rotor must be

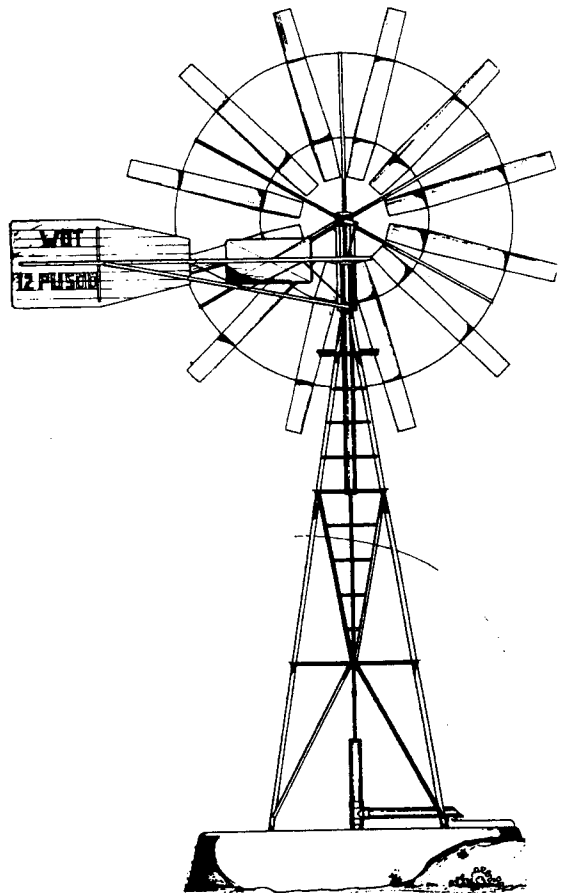put perpendicular to the wind by hand.

The pump is a single-acting piston pump which consists of a
piston and two valves, one valve in the piston (pistonvalve)
and one lower valve (footvalve), see lit. 1. The pump has
airchambers to smooth the flow and to reduce the shock forces
in the pumprod.

The total weight of the windmill is 400 kg and the material
costs in the Netherlands are about US$ 600,-.

For more information about the 12PU500 see lit.2 and 3.

1.2 The evaluation of the 12PU500 windmill

In 1986 the WOT evaluated the 12PU500 with help of reports of
T. Meyer, Rakish and Majithia from India and J. Keuper from
Indonesia and with help of the experiences on the testfield of
the WOT. Especially the evaluation report of 155 installed
12PU500 windmills by mr. Rakish is very detailed and accurate
and therefore helpful.

The conclusions of the evaluation (from mr. Rakish) were:
- Feasibility studies are very important before starting to
  introduce windmills.
- The quality of the iron profiles is often bad in developing
  countries. Pipes aren't circular, angle-irons aren't
  straight, the strength of the iron materials is small, etc.
- The axis of the rotor isn't horizontal but a little bit
  sloping (70% of the windmills).
- The blade supports (100% of the windmills) and the blade
  tips (30%) are cracked.
- There is too much or to little play between the head and the
  towerpipe (70%).
- Sometimes high windspeeds lift the head and rotor from the
  towerpipe.
- The safety system is wrong constructed or installed (80%),
  the teeth of the toothed handle wear out too much.
- The transmission: the crank is welded obliquely
  the crankpin is bolted obliquely to the crank
  the crank bearing and the crosshead bearing have too much

play

the wooden crosshead swells because of rainwater and sticks
inside the towerpipe

the crosshead wears out because the towerpipe is unround or
has a rough inner wall

the crank is not removable

the crankholes are eccentric.
- The pump: the airchambers leak often by leaking welds or
  washers

the washers dry up and start to leak

the bronze bushes for guiding the pumprod wear out

the wooden piston swells in the water and sticks inside the
pumpcylinder

the piston wears out too fast

bad axial alignment of the pumpcylinder and the rising main.

Other conclusions of the evaluation:
- The tower is difficult to transport.
- The tower is too low.
- The pumprod bents and rubs against the rising main.
- The safety system must be full-automatic. That means that
  the rotor turns back in the wind after the storm.
- Nylon for the bearings and sesame-wood for the cross-head is
  difficult to obtain.
- The air escapes slowly out of the airchambers. when the
  airchambers are full of water the shock forces in the pump
  rod increase much.
- The yaw bearing (head-towerpipe) is iron on iron. It
  wears out when it has no frequent lubrification.
- The windmill is too expensive for poor farmers with less
  than 1,25 ha. land.
- There is no interest of the local people when there isn't
  sufficient guidance and participation.

1.3 The requirement list

With help of the results of the evaluation a new list of
requirements is made:

General: - long-life, for replaceable parts a minimum life of
            two years, for the rest minimum of fifteen years
         - the whole windmill transportable on a roofrack of a
            cross country car like landrover.

Materials: - an efficient use
           - easily available
           - not too expensive.

Manufacturing: - possible in a simple workshop (electric
                welding appliance, drill machine, a simple
                lathe and hand tools)
               - the dimensional accuracy of the design not
                too high.

Rotor: - a lower tipspeed ratio
       - not too heavy so that it is possible to pull it up by
         hand (not heavier than the rotor of the 12PU500)
       - stronger blade tips and supports
       - high power-coefficient C_p
       - high starting torque (starting windspeed v_{start} low).

Safety system: - limiting the axial forces on the rotor
               - limiting the rotation speed of the rotor
               - full-automatic
               - 100% dependable.

Head: - avoiding that the head lifts from the tower
      - the yaw bearing not iron on iron.

Transmission: - avoiding high shock forces in the pumprod
              - no wooden cross-head in the towerpipe
              - an alternative for nylon bearings
              - a crank which is strong enough and easy to
                make the crankpin parallel to the rotor
                shaft)
              - the construction strong enough with a fatigue
                load.

Pump: - simple construction (no airchambers)
      - the piston not made of wood
      - no guiding of the pumprod
      - the construction strong enough with a fatigue load
      - pumpcylinder and rising main easily to align

        - high efficiency.

Tower: - higher than 6 meters
        - construction strong enough for a windspeed of 12 m/s
          and the rotor perpendicular in the wind and for a
          windspeed of 40 m/s and the rotor out of the wind
        - possible to install by hand.

The redesign of the 12PU500 is done by a group of ten people, see foreword. The windmill design was separated in five parts, namely:
- the rotor
- the safety system and head construction
- the transmission
- the pump
- the tower
(see next chapters)

CHAPTER 2. THE ROTOR

2.1. The rotor dimensions

The code 12PU500 means that the rotor has a diameter of 500 cm
and 12 blades and is coupled to a pump unit.
The blades of the 12PU500 windmill are made of steel sheet of
1 mm thick and have a length of 2 m. One of the conclusions of
the evaluation was that the blades are too long. At storms the
tips and the inner part can collapse and crack. To avoid this
we can make a third ring to support the blades but this will
make the construction too heavy and more expensive. A better
solution is to choose the length of the blades shorter. The
standard measurements of steel sheet are 1x2 m. We chose a new
blade length of 1 m so that the blades can be cut out of the
steel sheet efficiently. With the same construction of the
rings and spokes the rotor diameter became smaller, namely 4,5
m (450 rotor). A shorter length of the blades has the
advantage that a smaller sheet roller can be used to make the
bent blade profile. The disadvantage is the loss of power
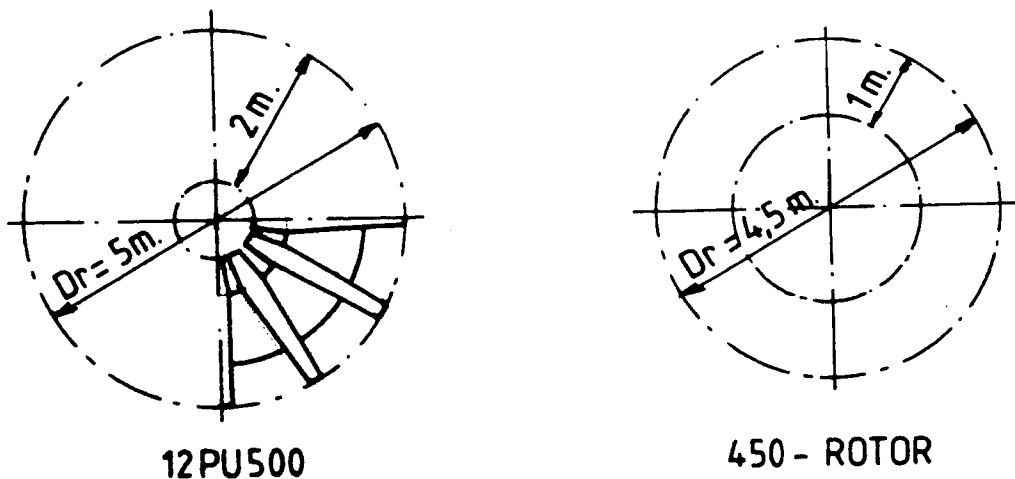because of the 'hole' in the center of the rotor.

figure 2.1: the 12PU500 and 450 rotor

2.2. The choice of the design tipspeed ratio

The design tipspeed ratio (λ_d) is the tipspeed ratio at the design windspeed. At this windspeed the overall efficiency of the windpump is maximum. The design tipspeed ratio of the 12PU500 is two. For the 450-rotor we chose a lower design tipspeed ratio, namely:

$$\lambda_d = 1.$$

We wanted to have a pump without airchambers. But these pumps have high peakforces in the pumprod at high rotation speeds. We thought that the peakforces are lower when we choose a lower design tipspeed ratio. The conclusion of appendix 2 is that a lower design tipspeed ratio does not result in lower peakforces for the windspeeds of 9 m/s (fatigue load) and 12 m/s (maximum 'static' load). The peakforces are more or less equal for $\lambda_d = 1$ and $\lambda_d = 2$.

The advantages of a lower design tipspeed ratio are:
- less wear of the piston and the pump cylinder
- lower gyroscopic moment of the rotor

The design windspeed of the 12PU500 is 3 m/s. We took the same design windspeed for the new design, so:

$$v_d = 3 \text{ m/s}$$

2.3. The blades

The blades are made of steel sheet of 1 mm. The sheets are bent as in figure 2.2. with $f/c = 0,1$. From table 2.1 of lit.4 we find the C_d/C_l ratio= 0,02. The number of blades (B) must be a multiple of 6 because of the 6 spokes: B= 6, 12, 18 or 24.

figure 2.2: the blade profile

To have a high starting torque the solidity (= area of all the
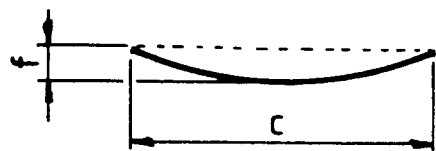
blades divided by the rotor area) must be high too (lit.4). Therefore it is better to have 18 or 24 blades in stead of 6 or 12. A rotor of 24 blades would be very heavy, so we chose 18 blades:

B= 18
D_r= 450 cm
} 18PU450 rotor

To design the blade dimensions we need the following data:
- the rotor radius (R_r)
- number of blades (B)
- design tipspeed ratio (λ_d)
- lift coefficient (C_l), see table 2.1, lit.4
- corresponding angle of attack (α)

and 4 formulas, see lit.4:

1. $\lambda_r = \lambda_d * r / R_r$ (formula 2.1)

2. $\phi = {}^1/_3 * \arctan 1/\lambda_r$ (formula 2.2)

3. corde $c = \dfrac{8*\pi*r}{B*C_l} *(1-\cos\phi)$ (formula 2.3)

4. $\beta = \phi - \alpha$ (formula 2.4)
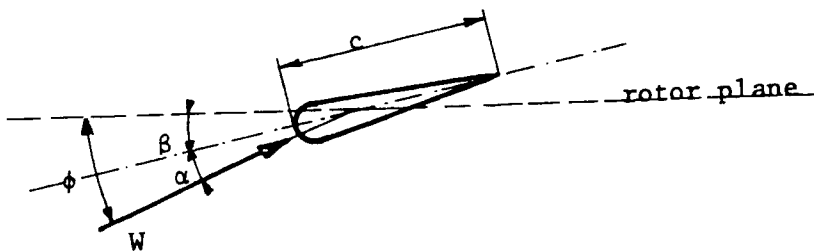
figure 2.3: blade setting β

For the 18PU450:
R_r= 2,25 m, C_l= 1,25, λ_d= 1, B= 18 and α= 3°

The corde (c) from r= 1,25 till r= R_r= 2,25 is 0,34-0,35 m. We chose c larger, c=0,39 m, because we want a high solidity compensating the hole in the center. A high solidity gives a high starting torque. The angle β for the support on the inner ring is 38° and for the support on the outer ring 30°.
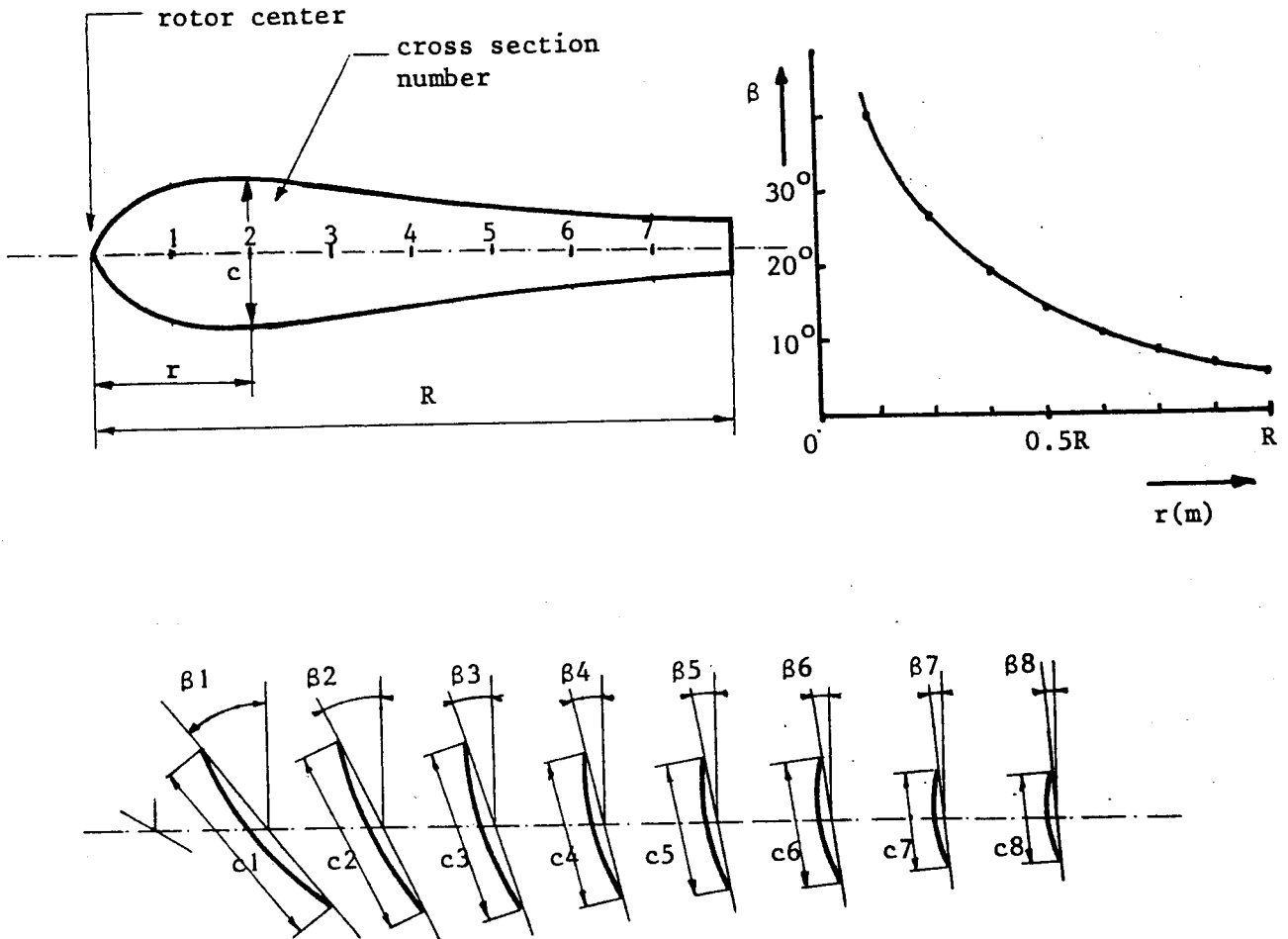
figure 2.4: blade form, twist and cross sections

position	r in m	λ_r	ϕ in °	β in °	c in m
1	0,28	0,125	55,2	52,2	0,13
2	0,56	0,25	50,6	47,6	0,23
3	0,84	0,375	46,3	43,6	0,29
4	1,13	0,5	42,1	39,3	0,33
5	1.41	0,625	38,7	35,7	0,35
6	1,69	0,75	35,4	32,4	0,35
7	1,97	0,875	32,5	29,5	0,34
8	2,25	1	30	27	0,34
support innerring	1,25	0,56	40,6	37,6	0,34
support outerring	1,90	0,84	33,2	30,2	0,35

The blade supports of the 12PU500 are made of steel sheet of 1 and 2 mm. The manufacturing of these supports requires much labour. The evaluation showed us that the supports can crack. Therefore we chose flat iron 30x6 mm for the supports of the 18PU450, see drawing nr. 2. This construction is easier to make and stronger. The disadvantage is that it is a little bit heavier.

2.4. The power and torque coefficient

The mechanical power which is converted by the rotor from the windpower is:

$$P_r = C_p * \tfrac{1}{2} * \rho_a * v^3 * A_r \qquad \text{(formula 2.5)}$$

The power coefficient C_p of the 12PU500 is measured in a windtunnel: $C_p = 0,32$. The C_p-value of the 18PU450 is not measured in a windtunnel but with lit.4 we can predict it:

$C_d/C_l = 0,02$
$\lambda_d = 1$
$B = 18$
$\left. \right\} \quad C_{p,th}$ (=theoretical C_p-value) $= 0,39$

When we calculate the losses of the hole in the center of the rotor then the maximum power coefficient is (see lit. 16):

$$C_{p,\text{max}} = C_{p,th} * (2 * R_r * 1 - 1^2)/R_r^2 = 0,27$$

The torque (Q) characteristic of the rotor and the load is very important. A piston pump has an almost constant torque which results in a relatively high starting torque (Q_{start}). Therefore it is important that the rotor has a high starting torque too.

The torque is given by:

$$Q = C_Q * \tfrac{1}{2} * \rho_a * v^2 * A_r * R_r \qquad \text{(formula 2.6)}$$

The torque coefficient $C_Q = C_p/\lambda$ (formula 2.7)

The starting torque coefficient is determined by the empiric relation: $C_{Qstart} = 0,5/\lambda^2$ (formula 2.8)

The 12PU500 ($\lambda=2$): $C_{Qstart} = 0,5/2^2 = 0,125$

The 18PU450 ($\lambda=1$): $C_{Qstart} = 0,5/1^2 = 0,5$

From experiences on rotors with $\lambda=1$ it appeared that this value of 0,5 is too high. According to lit. 16 the starting torque coefficient is:

$C_{Qstart} = 0,75*B*(R-\frac{1}{2}*1)*C_l*c*1/(\pi*R_r^3) = 0,28$

(with $C_l = 1,1$ for $\alpha = 90°-\beta \approx 54°$)

The starting windspeed (v_{start}) of the 12PU500 is more or less 2,5 m/s. The starting torque at this windspeed is;

12PU500: $Q_{start} = C_{Qstart}*\frac{1}{2}*\rho_a*v_{start}^2*A_r*R_r =$

$0,125*\frac{1}{2}*1,2*2,5^2*18,8*2,5 = 21,2$ Nm

18PU450: $Q_{start} = 0,28*\frac{1}{2}*1,2*2,5^2*11*2,25 = 26,0$ Nm

The 18PU450 has a higher starting torque.

2.5. Measurements of the windspeed-rotationspeed relation

The 18PU450 prototype on the testfield of the WOT is equipped with an anemometer beside the rotor and a rotation speed meter on the rotorshaft. The two meters are connected to a computer which registers the data. The anemometer is a cup-anemometer with a reed-contact on the shaft. The registered cycle time is a measure for the windspeed. The anemometer is calibrated w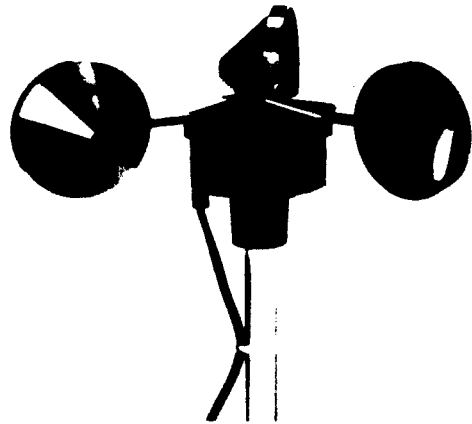ith another anemometer, an Aeolian Kinetics Wind Prospector 4000. The computer registers once every minute the cycle time of the anemometer and of the rotor. A computer program calculates the windspeed, the rotation speed of the rotor and the tipspeed ratio. It calculates also the average tipspeed ratio for

figure 2.5: cup-anemometer

intervals of windspeed. measurements are done with and without
load. The load is a 4" pump with D_p= 105 mm and R_c= 120 mm. The
waterlifting head is 10,5 m. The average tipspeed ratio at v=
3 m/s is 1,0 and at v= 9 m/s 1,6, see fig. 2.6. The average
tipspeed ratio without load is 1,7. The rotor stood
perpendicular in the wind during the measurements (the
prototype has the hysteresis safety system, see par. 3.2). The
measurements aren't accurate because the windspeed is
fluctuating very much. For more accurate measurements we have
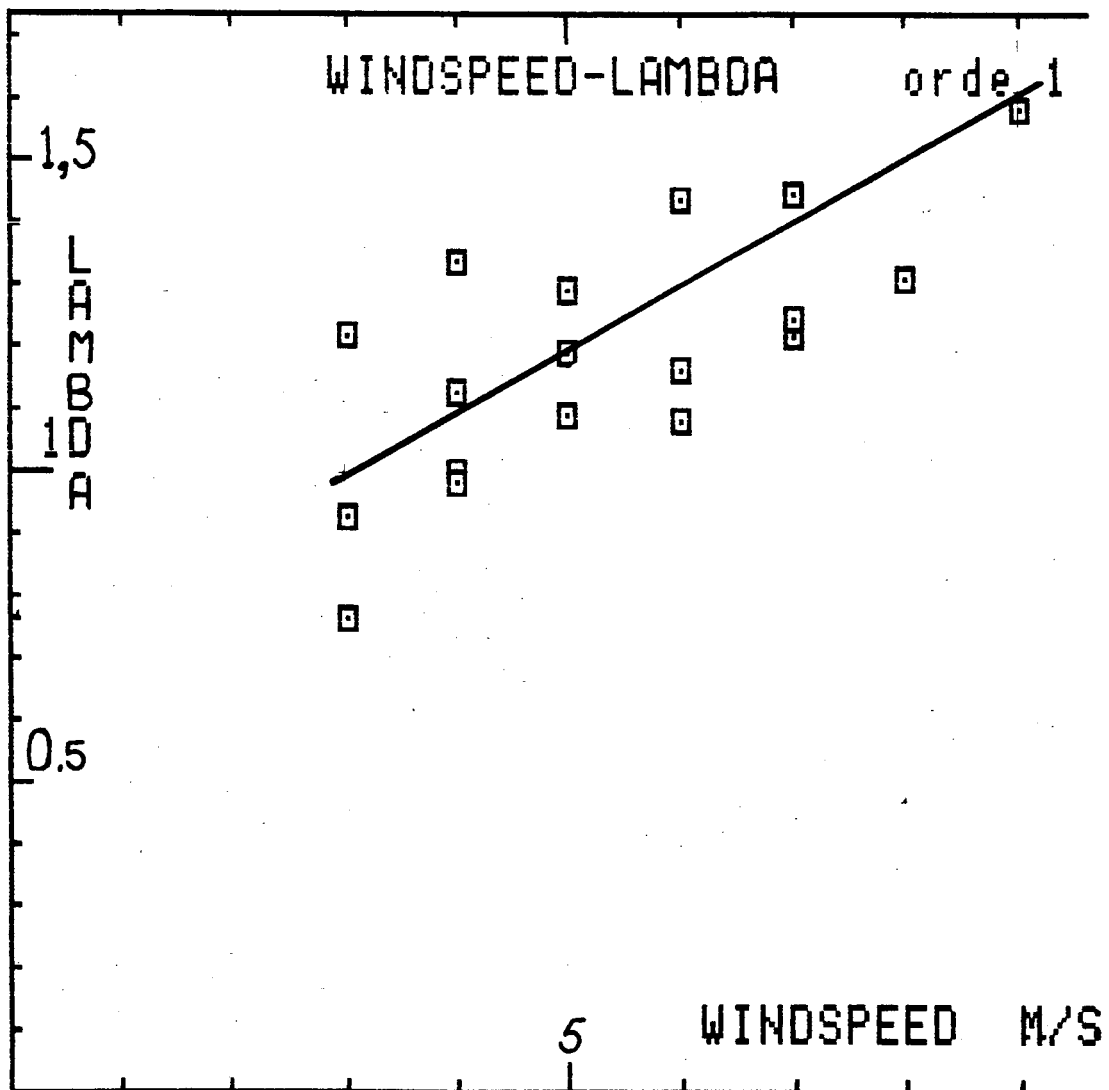to use a windtunnel with constant windspeeds.

figure 2.6: tipspeed ratio as function of the windspeed with
load

CHAPTER 3. THE SAFETY SYSTEM AND THE HEAD CONSTRUCTION

3.1. General

The safety system has to limit the axial windforces on the rotor and the rotation speed of the rotor.

The 12PU500 has a half-automatic safety system. When the windspeed is too high the rotor turns out of the wind. After the hard winds the rotor must be put in the wind again by hand.

The 18PU450 has a full-automatic system, without necessary human interference. Before designing the system all kind of existed systems for windpumps were evaluated, see lit.6.

3.2. The hysteresis system

The first choice was a completely new system, the so-called hysteresis system, see fig.3.1.

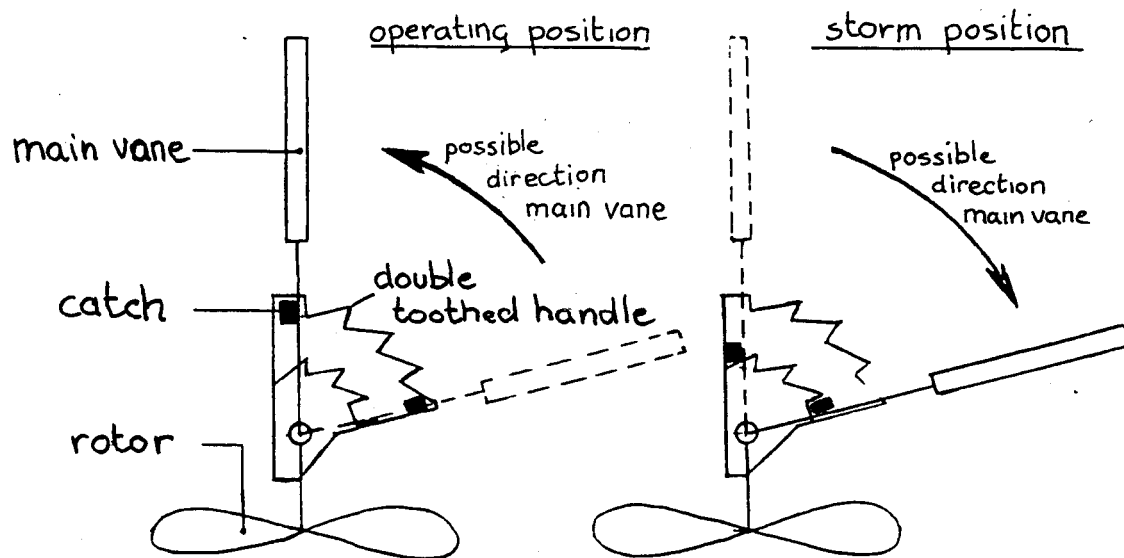figure 3.1: the safety mechanism with a double toothed handle

This mechanism with a double toothed handle has a comparable working as the half-automatic system of the 12PU500. Only this mechanism has a second toothed handle which takes care that the rotor turns in the wind again. In the normal position the catch is in the toothed handle A. The catch is the extension of the arm of the help vane as in fig.3.2 shown. When the

windspeed becomes too high
the windpressure on the help
vane pushes the help vane
backwards. The catch falls in
the toothed handle B. The
head with the rotor is
unlocked from the tail and
will turn out of the wind.
This turning starts because
of variations in the wind
direction and then continues
because the windforce on the
rotor is eccentric to the
towerpipe axis and causes a
torque. When the windspeed

figure 3.2: the double toothed
handle with the help vane

decreases the help vane falls down in its normal position by
its weight. The catch is in toothed handle A again. Because of
variations in the wind direction the rotor will turn in the
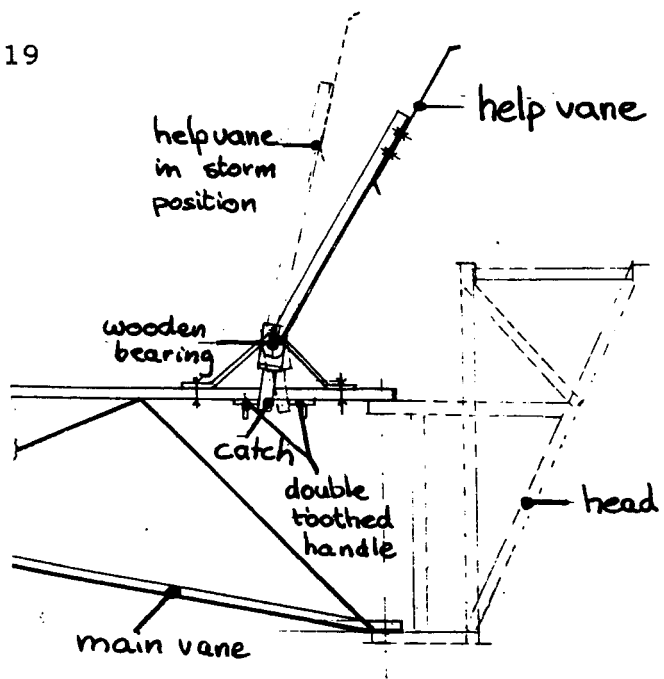wind again, tooth after tooth.

The help vane works concerning a
hysteresis. The relation between the
windspeed and the angle of the arm and
the horizontal (δ) is given in fig.3.3.
At v_{max} (12 m/s) the help vane is blown
backwards and the rotor starts to turn
out of the wind. At v_l (more or less 7,5
m/s) the helpvane falls forwards in its
normal position and the rotor starts to
turn in the wind again.
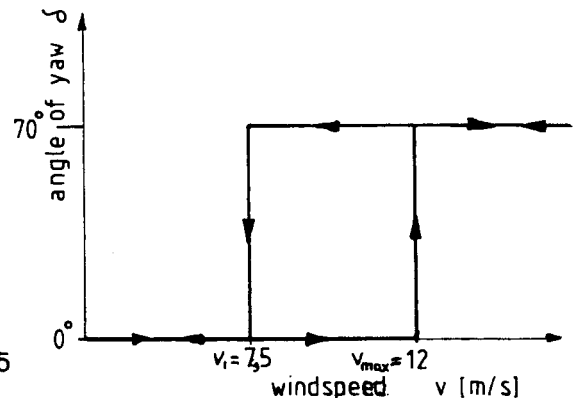The graph of fig.3.3 shows an hysteresis
loop.

figure 3.3:
relation between
the windspeed v and
angle δ

The 18PU450 prototype is equipped by the hysteresis system.
Till now the system has done its duty, but there appeared some
problems:
- The time that the rotor needed for turning in the wind again
  is sometimes more than one day, especially when the

windspeed decreased lower than 4-5 m/s. This causes a loss
in the output.
- Sometimes the rotor didn't turn out of the wind directly
  after the catch fell in toothed handle B but reacted just
  after 4-5 seconds. During this time the rotation speed can
  increase too much and damage the pump.
- When the rotor was turning out of the wind the shock of the
  catch against the end of the toothed handle was rather heavy
  sometimes.

Other disadvantages of the hysteresis system are:
- The design depends on variations of wind direction. In areas
  where the wind direction is rather constant the system
  possibly may not function well.
- The construction of the help vane has a bearing which never
  may have too much friction.
- The manufacturing of the double toothed handle requires much
  labour.

These points make clear that the hysteresis system isn't
dependable for 100% and can cause output losses. We chose
another system, namely the well-known inclined hinged vane
safety system.

3.3. The inclined hinged vane system

This system has a tail which doesn't hinge in an horizontal
plane but an inclined plane, see fig.3.4.
When the windspeed increases the thrust on the side vane and
on the rotor by the wind increases too. The thrust causes a
higher moment which turns the rotor out of the wind. The tail
vane will not turn so far because the wind thrusts the tail
vane in the wind direction. This position of the tail vane is
not its lowest position, so its weight will give a moment
around the s-axis of the hinge. The tail vane will make an
angle (α) with the winddirection. The force caused by the wind
thrust on the tail vane (F_v) will give a moment around the z-
axis of the towerpipe. This moment will stop the rotor from
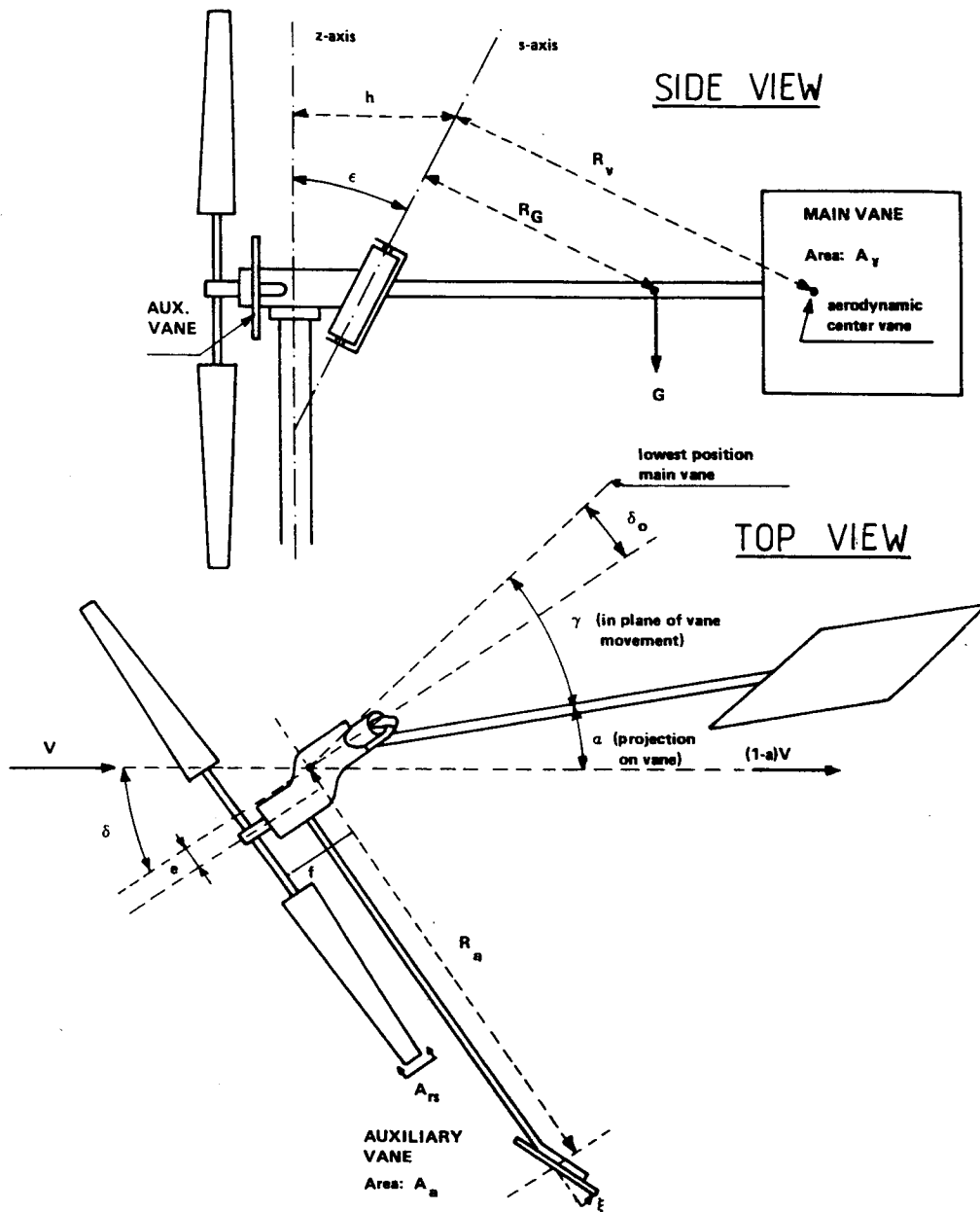
figure 3.4: the inclined hinged vane safety system

turning out of the wind. At this point there is a labile balance between the position of the rotor and the tail vane. Between the rotor axis and the wind direction we find the angle δ. When the windspeed increases the angle δ increases too.

This system wasn't the first choice because:
- It is difficult to adjust
- There is a labile balance between the rotor and the tail vane. Only for one certain windspeed the rotor stands

perpendicular to the wind direction. This means a loss of
power.
- The wooden yaw bearing of the Diever 450 has more
  friction than ball-bearings or roller-bearings. This means
  an uncertain factor for the behaviour of the system.

The advantages of the system:
- Simple construction
- Not sensitive for failure
- Full-automatic
- Applicated on many other windmills.

After it was clear that the hysteresis system isn't the ideal
system we made a new investigation to other safety systems.
The conclusion was that the inclined hinged vane system was a
good alternative. The system can be partly adjusted by
variating the side vane area (A_{rs}) and arm length (R_a). The
angle of the hinge (ϵ) on the prototype should be variable
too. The influence of the wooden yawing bearing could only be
find out by trying on a prototype.

To predict the behaviour of the system (relation between the
windspeed and the angle δ) we can use the next formulas:
Forces:

thrust on rotor   : $F_{rt} = C_t * \frac{1}{2} * \rho_a * (v * \cos \delta)^2 * A_r$

side force on rotor   : $F_{rs} = C_f * \frac{1}{2} * \rho_a * (v * \sin \delta)^2 * A_{rs}$

wind normal force on tail vane   : $F_v = 2,6 * \alpha * \frac{1}{2} * \rho_a * (1-a)^2 * v^2 * A_v$

weight normal force on tailvane   : $F_{vg} = G * \sin \epsilon * \sin \tau$

wind normal force on side vane   : $F_a = C_N(90° - \delta - \xi) * \frac{1}{2} * \rho_a * v^2 * A_a * \cos \xi$

(formulas 3.1-3.5)

Moments:
around s-axis: $F_v * R_v = F_{vg} * R_g$ (formula 3.6)
around z-axis: $F_{rt} * e + F_{rs} * f + F_a * R_a = F_v * (h * \cos \gamma + R_v * \cos \epsilon)$

(formula 3.7)

The angles:
$\sin \alpha = \sin \delta * \cos \gamma - \cos \delta * \cos \epsilon * \sin \gamma$ (formula 3.8)
for small angles of α: α (in rad) $\approx \sin \alpha$ (formula 3.9)

The two moment balance equations around the z-axis and the s-axis can be combined in one formula:

$$C_t * \cos^2(\delta-\delta_0) * \frac{A_r * e}{A_v * R_v} + C_f * \sin^2(\delta-\delta_0) * \frac{A_{rs} * f}{A_v * R_v} +$$

$$C_N (90° - (\delta-\delta_0) - \xi) * \cos \xi * \frac{A_a * R_a}{A_v * R_v} =$$

$$2,6 * F * \frac{T}{\sqrt{(1+T^2)}} * (1-a)^2 * \left\{ \frac{h}{R_v * \sqrt{(1+T^2)}} + \cos \epsilon \right\} \qquad \text{(formula 3.10)}$$

with:

$$F = \frac{G * R_G * \sin \epsilon}{2,6 * \tfrac{1}{2} * \rho_a * (1-a)^2 * v^2 * A_v * R_v} \qquad \text{(formula 3.11)}$$

and:

$$T = \frac{\sin \delta}{F + \cos \delta * \cos \epsilon} \qquad \text{(formula 3.12)}$$

The solution of these equations should be the angle δ as a function of the windspeed v:

$$\delta = \delta(v) \qquad \text{(formula 3.13)}$$

In order to find this function we need to know a survey of parameters. For the Diever 450 these parameters are:

rotor	value	unit
D_r= rotor diameter	4,5	m
A_r= rotor area	15,9	m²
e= rotor eccentricity	0	m
A_{rs}= rotor side area= 0,075xA_r	1,2	m²
f= distance rotor plane to z-axis	0,8	m
side vane		
A_a= side vane area	0,42	m²
R_a= distance aerodynamic center of the side vane to the z-axis	3	m
ξ = side vane angle	0°	degree
main vane		
A_v= main vane area	2,75	m²
R_v= distance aerodynamic center of the main vane to the s-axis	3,37	m
δ_0= preset angle of the main vane	0°	degree
h= see fig. 3.4	0,35	m

	value	unit
ϵ= angle of inclination of the s-axis	10°	degree
G= main vane weight	550	N
R_G= distance gravity center to the s-axis	2,45	m

aerodynamic

ρ_a= air density	1,2	kg/m^3
C_f= rotor side force coefficient	$^8/_9$	
C_N= normal force coefficient	see fig. 3.5	
a= wake factor	see fig. 3.6	
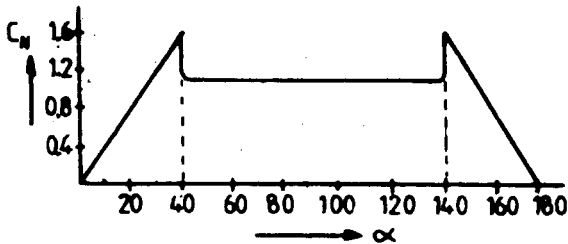
figure 3.5: normal force coefficient
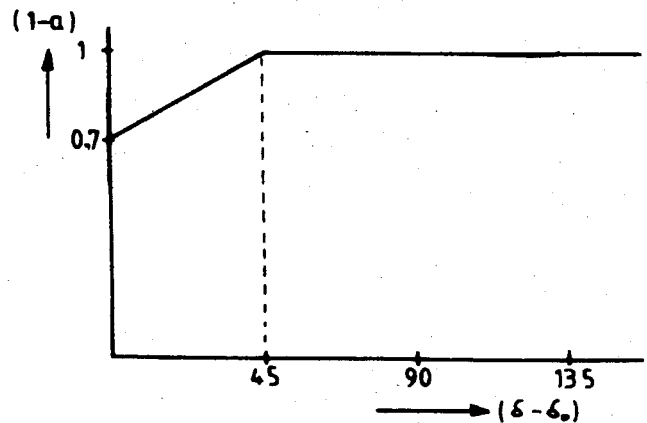
figure 3.6: the wake factor a

Now the theoretical δ-v curve for the behaviour of the safety system can be calculated, see fig. 3.7.

In fig. 3.7 we assume that the friction of the wooden yaw bearing is negligible. This is not true. The wooden bearing has a friction that causes a counteracting moment around the z-axis (M_f), with a value of more or less 40 Nm.
When we take this value of M_f= 40 Nm and calculate δ(v) for an increasing and a decreasing windspeed then we get the curves of figure 3.8.

For more information about inclined hinged vane systems see lit. 5 and 8.

figure 3.7: δ-v curve without friction of the yaw bearing

figure 3.8: δ-v curves for increasing and decreasing windspeed

3.4. The strength calculations of the safety system

For the Diever 450 we chose the inclined hinged vane system.
The dimensions are given in par. 3.3 and drawing nr.3.
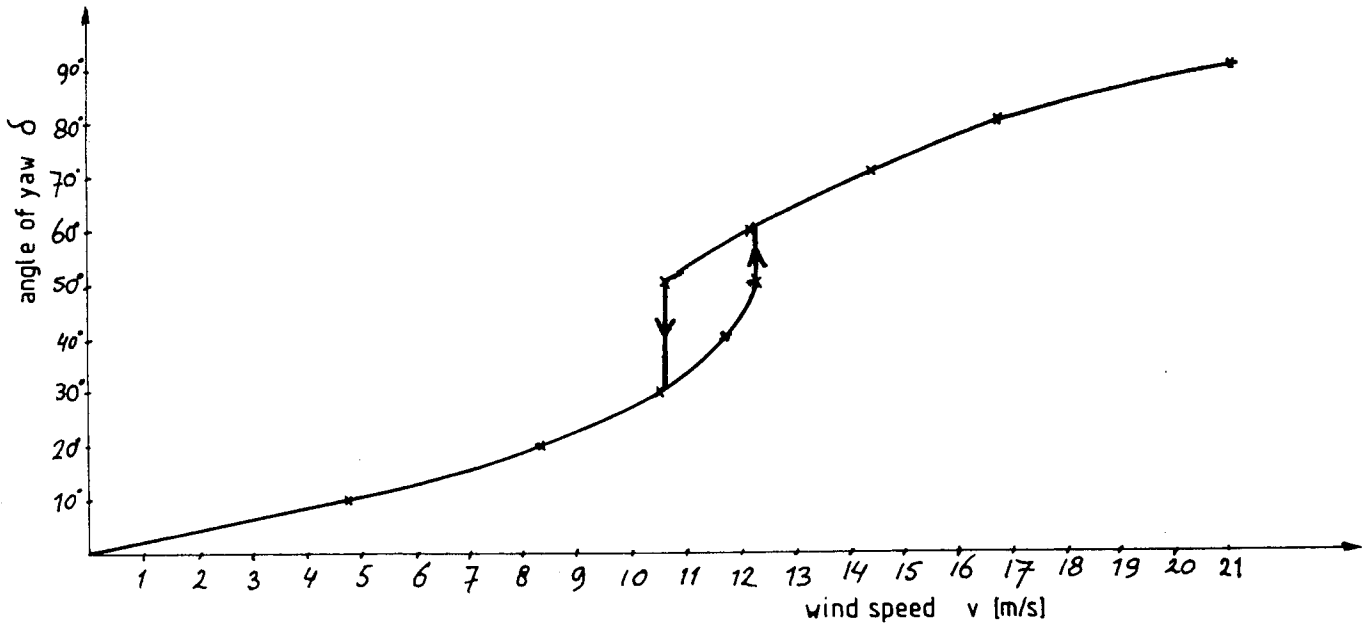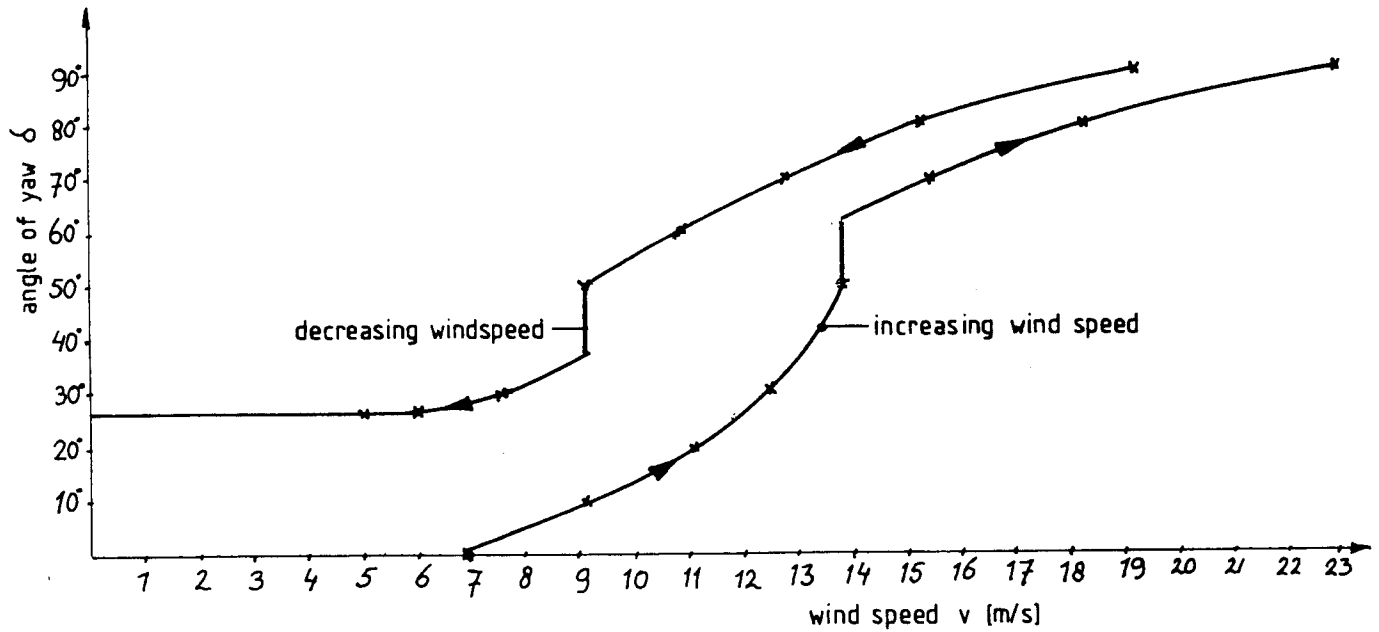The two hinges of the main vane are wooden bearings. The
admissible surface pressure for wood is 2 N/mm² for constant
load and 5 N/mm² for peakloads.
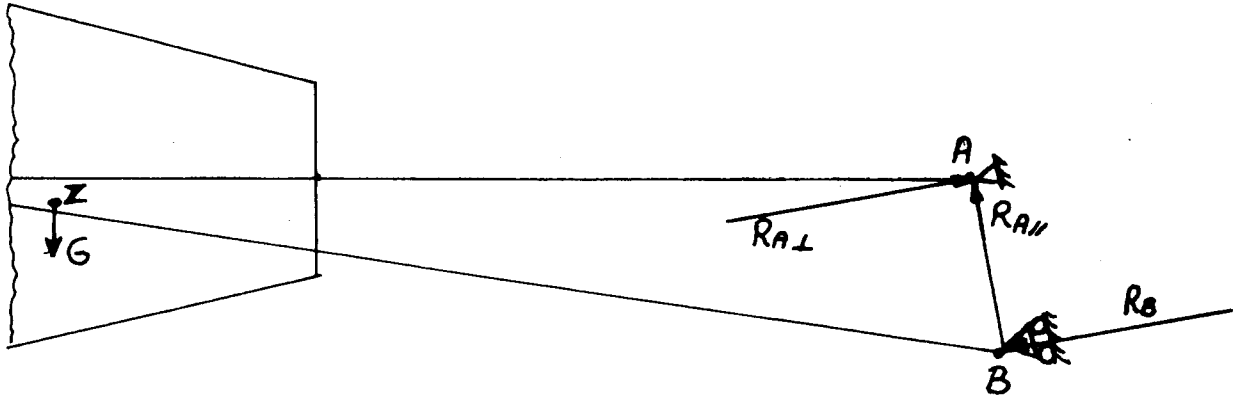
figure 3.9: reaction forces on the hings

$\Sigma V = 0$ $R_A = 3200$ N

$\Sigma H = 0$ $R_A = 560$ N

$\Sigma M_{around\ A} = 0$ $R_B = 3040$ N

The bearing A is most critical. The surface pressure:

in the axial direction: $\sigma_v = 0,55$ N/mm²

in the radial direction: $\sigma_v = 1,11$ N/mm²

The barforces in the main vane construction can be calculated
by the Cremona-method, see fig. 3.10.
The barforces are given in the next table:

bar nr.	profile	tensile	pressure	σ_t	σ_b
1	L40x40x4		3150		21,6
2	L40x40x4		2600		26,7
3	L40x40x4		2350		24,1
4	L40x40x4	2400		3,9	
5	L40x40x4	3000		4,9	
6	1½"	600		1,3	
7	L30x30x3	250		1,4	
8	L30x30x3		400		8,5
9	L30x30x3	300		1,7	

The forces in [N], the stresses in [N/mm²].
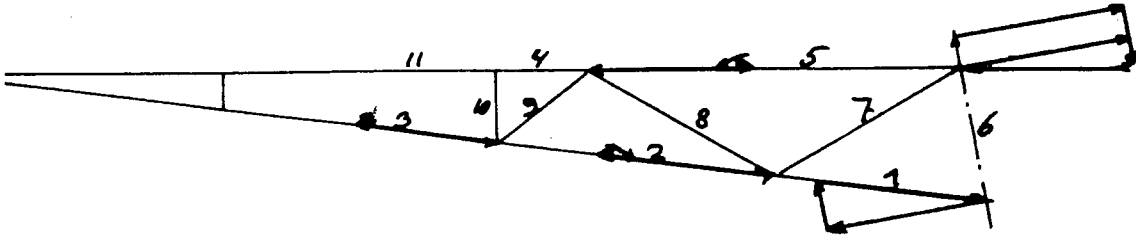
figure 3.10: barforces calculated by Cremona-method

The admissible stress is 160 N/mm². The stresses in the bars
are much lower, so the construction is strong enough. We could
make the construction lighter but these calculations are only
based on the load of the weight. Loads which are not included
in the calculations are:
- torsion caused by turbulences in the wind. These turbulences
  can cause different windpressures on the upperpart and on
  the lower part of the main vane. That will give a torsion on
  the construction.
- shock against the bumper. When the rotor turns out of the
  wind till the farthest position the main vane hits the
  bumper. The bumper avoids that the rotor hits the main vane.
  When we assume that the rotor turns out of the wind with an
  angular speed of 0,63 rad/s and hits the bumper, we can
  calculate the bending stress on the construction just where
  it hits the bumper: σ= 155 N/mm² (the admissible stress is
  160 N/mm²).

These points make clear that we cannot make the construction
lighter.

The forces on the side vane are:
- the weight of the vane and its arm construction (resp. 100 N
  and 140 N)
- the wind force on the vane (maximum 60 N)
The bars ADM and CD are made of $^3/_4$" gaspipe, bar BD of ½".
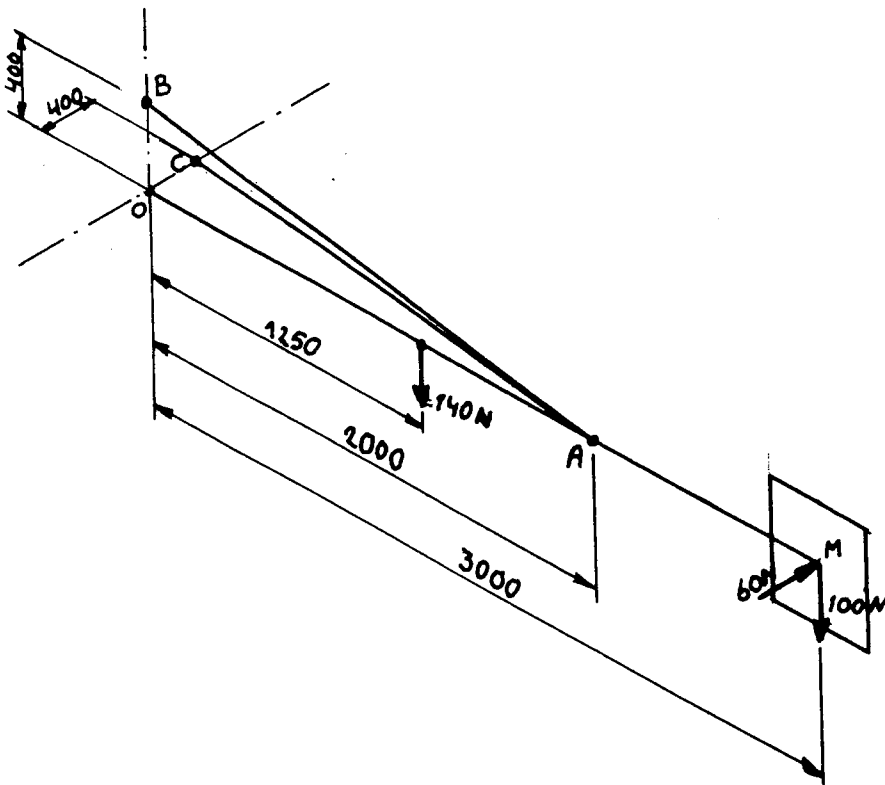
figure 3.11: forces on the side vane

The reaction forces in the connections A,B and C are calculated and given in fig. 3.11. The bar forces are:

bar AD: F_{AD}= -750 N (pressure)

bar BD: F_{BD}= +1220 N (tensile)

bar CD: F_{CD}= -460 N (pressure)

For $3/4$" gaspipe the admissible pressure force with a buckle length of 2 m is: $\bar{F}$= 3100 N

The bending moment next to D: M_D= 121 Nm. For $3/4$" gaspipe: σ= M/W= 109 N/mm².

We can conclude that the arm construction is strong enough.

MOVING PARTS OF -12PU500-WINDMILL

4

BRACKET PREVENTS THAT BEARINGHOUSE CRACKS OPEN!

CRANK FORCES

ROTOR WEIGHT

THRUST

SPOTWELDINGS

CLEARANCE THIS SIDE!

CLEARANCE THIS SIDE!

26 SLIDES IN CRANCK

SCRAP

WELDING HELPDEVICE

GENEROUS WELDINGS!

R=60 R=90 R=120

FITS SLIDELY INTO P.V.C. INSERT

Ø 40 x 45

Ø25

PUMPROD FITS HEAVILY

SLIDES EASILY INTO BOTTOM.END OF TOWERPIPE

PUMPROD, SLIDES EASILY

Ø 120

RECESS OF 0.2 MM

44 / 40

Ø25 (1") Ø20 (¾") Ø30 / Ø43

Ø 38 (½") 15 Ø1" / Ø33

33 Ø43 / Ø33

MEASUREMENT TO TOWERPIPE EDGE 670

SEAM

PUMPROD CONNECTION SEE DRAWING Nº6

TOWERPIPE P.V.C. INSERT

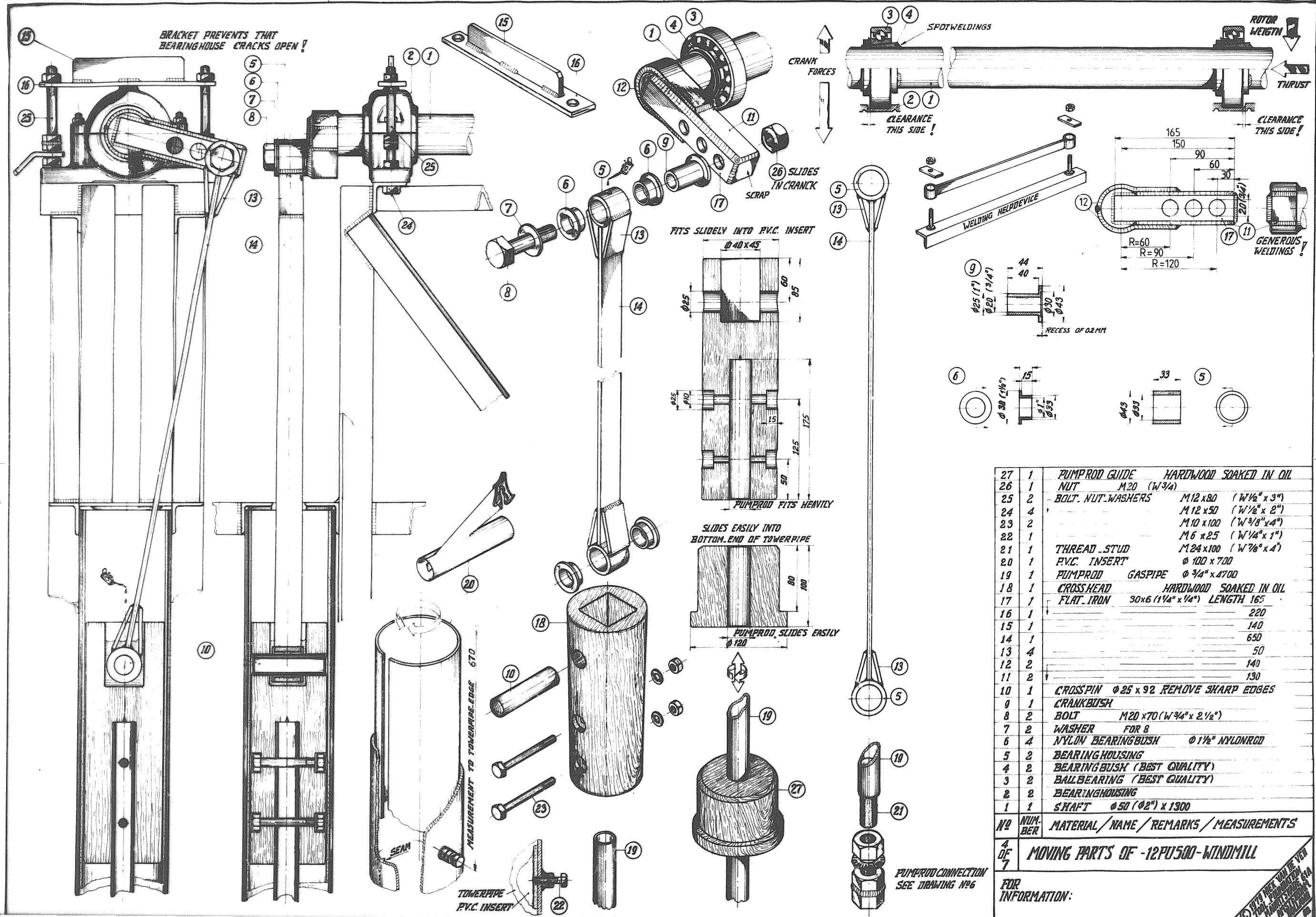| Nº	NUMBER	MATERIAL / NAME / REMARKS / MEASUREMENTS
27	1	PUMPROD GUIDE HARDWOOD SOAKED IN OIL
26	1	NUT M20 (W¾)
25	2	BOLT. NUT. WASHERS M12 x80 (W½" x 3")
24	4	M12 x50 (W½" x 2")
23	2	M10 x100 (W⅜"x4")
22	1	M6 x25 (W¼"x 1")
21	1	THREAD.STUD M24 x100 (W⅞"x 4")
20	1	P.V.C. INSERT Ø 100 x 700
19	1	PUMPROD GASPIPE Ø¾" x4700
18	1	CROSSHEAD HARDWOOD SOAKED IN OIL
17	1	FLAT.IRON 30x6 (1¼" x ¼") LENGTH 165
16	1	220
15	1	140
14	1	650
13	4	50
12	2	140
11	2	130
10	1	CROSSPIN Ø25 x 92 REMOVE SHARP EDGES
9	1	CRANKBUSH
8	2	BOLT M20 x70 (W¾"x 2½")
7	2	WASHER FOR 8
6	4	NYLON BEARING BUSH Ø 1½" NYLONROD
5	2	BEARING HOUSING
4	2	BEARING BUSH (BEST QUALITY)
3	2	BALL BEARING (BEST QUALITY)
2	2	BEARINGHOUSING
1	1	SHAFT Ø 50 (Ø2") x 1300

4 OF 7

FOR INFORMATION:

CHAPTER 4. THE TRANSMISSION
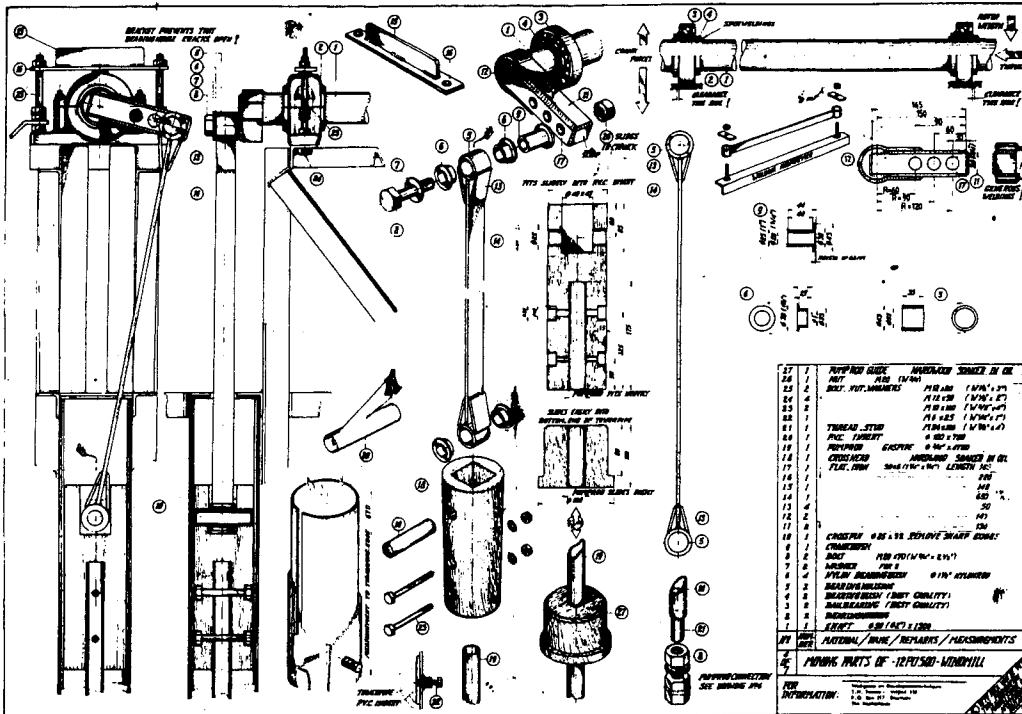
4.1. The mechanism

figure 4.1: the transmission of the 12PU500

The transmission of the 12PU500 consists of a crank-connecting rod- crosshead construction (see fig. 4.1). The evaluation in par. 1.2 gives a summary of the problems of the transmission. The requirement list of par. 1.3 is based on the results of the evaluation. One of the requirements is to avoid high peak forces in the pumprod. This can be achieved by:

- decreasing the acceleration of the piston at the moment that it passes its lowest position (at this moment the peak force appears).
- decreasing the acceleration of the water in the rising main by decreasing the ratio of the areas of the piston and of the rising main (A_p/A_{rm}).
- using an airchamber.

In chapter 5 these points are more detailed. By designing the mechanism of the transmission we can decrease the acceleration of the piston. The transmission of the Diever 450 as shown in

fig. 4.2 is a mechanism of four bars. At a constant angular speed of the rotor the upward movement of the piston takes more time than downward (202° of 360° rotation upward and 158° downward).

In appendix 2 the position of the piston (s) as function of the time (t) is calculated and given in graph nr. 1. By numeric differentiating we get the speed of the piston (v_p) as function of t (graph nr. 2) and secondly numeric differentiating we get the acceleration of the piston (a_p) as function of t (graph nr. 3). From the graph nr. 3 we see that the acceleration of the piston in its lowest position is:
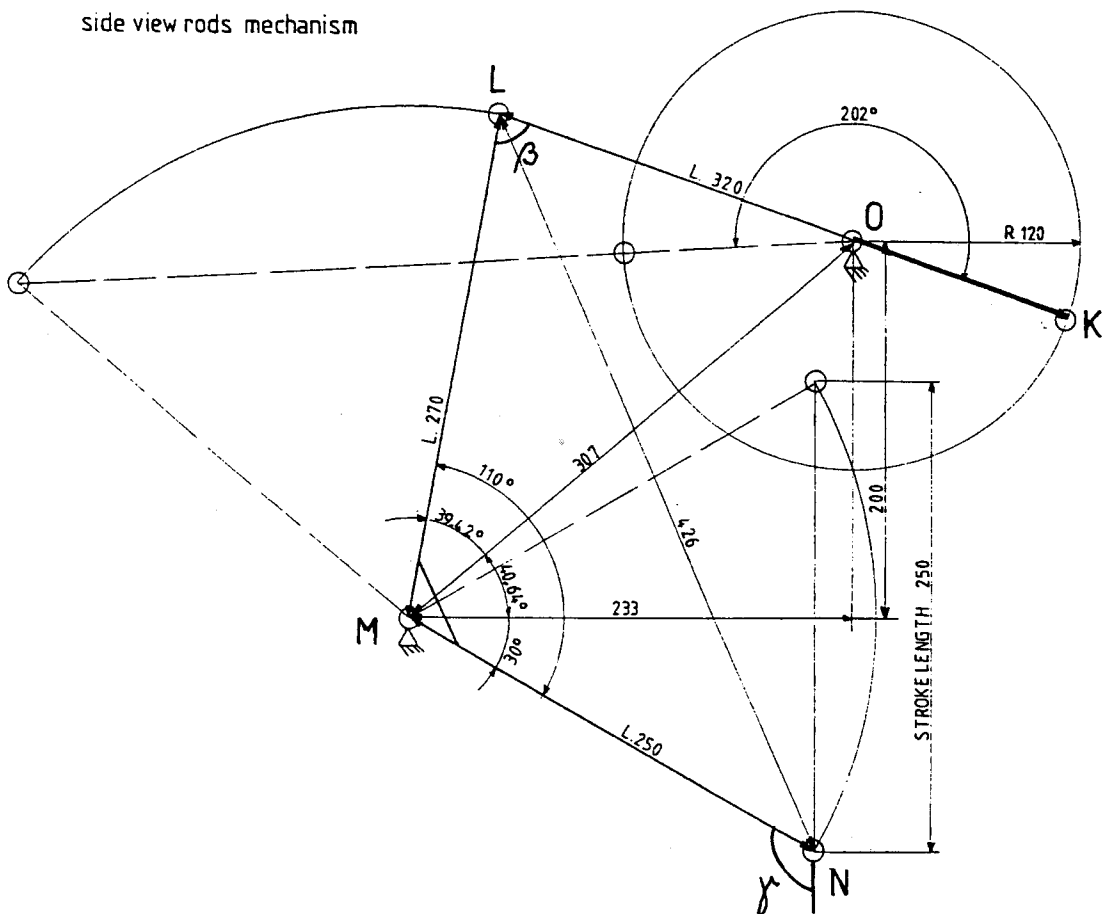
figure 4.2: four bars mechanism

a_p= 2,4 m/s² for Ω= 2π rad/s and s= 250 mm
With the same values of the angular speed and stroke length the transmission of the 12PU500 gives:
a_p= 4,7 m/s²!
The a_p with the transmission of the Diever 450 is **51%** of the a_p with the transmission of the 12PU500 and a crank length of 120

mm and **76%** with a crank length of 60 mm.

4.2 The construction and the strength

The construction of the transmission of the Diever 450 is
given in drawing nr.4. The crankpin is an iron shaft of ϕ30
mm. Flat irons of 30x6 mm are welded between the crankpin and
the rotorshaft. The crank length is 120 mm and is not
variable. A construction of a crank with a variable crank
length is given in appendix 6. From the crank a bar of 1"
gaspipe (the connecting rod) is connected to a tumbler
construction, 'the elbow', which is made of two iron sheets of
4mm. The pumprod is connected at the other side of the elbow.

All the bearing bushes are made of wood. The admissible
surface stress (σ_v) is 5 N/mm² for peak load. The projected
area (A_{proj}) of the bearings L, M and N is 27*70= 1890 mm², of K
is 30*70= 2100 mm². The admissible force on the bearings L, M
and N: $\bar{F}$= 5*1890= 9450 N.
The admissible force on the bearings in K: $\bar{F}$= 5*30*70= 10500 N
The peak force in the pumprod (F_{peak}) appears when the piston
passes its lowest position. With the piston in its lowest
position and the crank length of 120 mm the angle β= 81° and
γ= 120° (see fig. 4.2). The reaction forces in the bearings
are:
in the crank bearing K: R_K= 0,81*F_{peak}
in bearing L   : R_L= 0,81*F_{peak}
in bearing M   : R_M= 1,07*F_{peak}
in bearing N   : R_N= F_{peak}
The reaction force in M is the highest. If R_M= F= 9450 N then:
→ $\bar{F}_{peak}$= 8830 N

The pump makes a lot of strokes, possibly more than 10
million. It means that the load (the pumprod forces) is a
fatigue load. The admissible stresses for the iron parts with
a fatigue load are lower than with a static load. Therefore we
have to calculate the fatigue strength of the iron parts.

The fatigue strength of the crank:
The admissible stress is:
$\bar{\sigma}_e = \sqrt{(\sigma^2 + \tau^2)} = 105$ N/mm²
(for $\sigma_{min}/\sigma_{max} = 0$, see lit.11)
$\sigma = M/W = R_K * 44/(\pi/16 * 30^3)$
$\tau = R_K/A = R_K/(\pi/4 * 30^2)$
$\rightarrow \bar{R}_K = 8400$ N $\rightarrow \bar{F}_{peak} = 10370$ N

The connecting rod is loaded by a pressure force during the peak force in the pumprod:
$F = 0,81 * F_{peak}$
The buckle length is: $l_k = 320$ mm
$\rightarrow \bar{F} = 30520$ N $\rightarrow \bar{F}_{peak} = 37680$ N

The strength of the rotorshaft bearings:
The admissible reaction force in the bearing next to the crank is:
$\bar{R} = \bar{\sigma}_V * A_{proj} = 5*50*70 = 17500$ N
$F_{peak} \approx 0,89 * R$
$\rightarrow \bar{F}_{peak} = 15630$ N

The conclusion of these strength calculations is that for the transmission the admissible pumprod force is:

$\bar{F}_{pr} = \bar{F}_{peak} = 8830$ N

In stead of wooden bearings we can use ball-bearings when they are available. In appendix 5 there is the drawing of the transmission with ball-bearings.

CHAPTER 5. THE PUMP

5.1. General

The Diever 450 is coupled to a single-acting piston pump, as
the 12PU500. The piston pump has a piston which moves up- and
downwards (reciprocating) in a cylinder (pump cylinder). The
pump has two valves, the piston has a valve (piston valve) and
a second valve at the bottom of the pump (foot valve), see
lit. 1.

5.2. The pumprod forces

The pump of the 12PU500 has airchambers. The function of
airchambers is to smooth the pulsating flow of the pump and to
reduce the dynamic forces. But we experienced that the
airchambers were loosing their air slowly. There are four
possibilities for solving this problem:
- a flexible wall separating the air from the water (for
  example a rubber hose)
- an airpump filling the airchamber
- airchambers with a sniffle valve
- without airchambers and the pumprod forces lower than the
  admissible forces of the construction of the pump, pumprod
  and transmission.
Without airchambers the pump construction is more simple. But
it means also that the design windspeed is limited. A high
design windspeed means a large swept volume and causes high
pumprod forces (peakforces).

The peak forces in the pumprod appear just after the piston
has passed its lowest position. The peak forces consist of
different forces (see appendix 2):

$F_{peak} = F_{stat} + F_a + F_s + F_v + F_{vl}$

F_{stat} = force caused by the static pressure

F_a = force caused by the acceleration of the water, piston and
pumprod

F_s= force caused by the shock of the piston against the water column at the moment that the piston valve closes

F_w= force caused by the friction of the piston and the cylinder

F_{wi}= force caused by the conductor resistance in the pump and rising main (see lit. 9)

With the transmission of the Diever 450 the acceleration of the piston in its lowest position (a_p) is 51% of the a_p with the transmission of the 12PU500 at the same rotation speed, see chapter 4 (for R_c= 120 mm). Therefore the acceleration force F_a is much smaller.

The shock force F_s appears when the piston valve closes and the piston bumps against the standing water column. For the Diever 450 F_s is smaller than for the 12PU500 because the speed of the piston (v_p) of the Diever 450 at the moment of bumping is lower (71% of v_p of the 12PU500) at the same rotation speed.

It may be clear that it isn't easy to calculate F_{peak} this way. Especially the shockforce is difficult to determine. The C.W.D. did a lot of research on this matter. They made a program for the calculation of the pumprod forces of their pumps. They have found also another simple way to calculate it by using the next formula (lit. 10):

$$F_{peak}= k*\rho_w*H*A_p*(g + \Omega^2*R_c*A_p/A_{rm}) \qquad \text{(formula 5.1)}$$

k is an overshoot factor.
To make the formula more accurate we have to bring in A_{pr}.

$$F_{peak}= k*\rho_w*H*(A_p-A_{pr})*\{g + \Omega^2*R_c*(A_p-A_{pr})/(A_{rm}-A_{pr})\} \qquad \text{(formula 5.2)}$$

In this report I will use this formula. With my measurement results I will calculate k too, see par. 4.3. The measurements are done with two pumps on the pump test rig and on the Diever 450 on testfield of the WOT. The efficiency of the pump installation is determined by the indicator diagram of the

pump, see par. 5.6.

5.3. The rotation speed

Because of the reciprocating piston the system experiences a lot of fluctuating load cycles and fatigue symptoms can appear. The admissible stresses in materials under a fatigue load is much lower than under a static load. For the fatigue strength calculations of the pump I take the rated windspeed of v= 9 m/s and for the static strength the maximum windspeed v= 12 m/s, for both cases the rotor perpendicular in the wind. The tipspeed ratio λ_d of the Diever 450 is 1 for v_d= 3 m/s. For higher windspeeds the tipspeed ratio is higher, for v= 9 m/s and v= 12 m/s the tipspeed ratio $\lambda \approx 1,6$.

$\lambda = \Omega * R_r / v = \pi * n * D_r / v$

for v= 9 m/s: n= 60 r.p.m.

for v= 12 m/s: n= 80 r.p.m.

5.4. The pump dimensions

To determine the optimal swept volume of the pump I use the next formula (lit. 4):

$P_{rotor} = P_{pump} / \eta_{tot}$

$\frac{1}{2} * C_p * \rho_a * v^3 * \frac{1}{4} * \pi * D_r^2 = \rho_w * g * H * \Phi_{eff} / \eta_{tot}$ (formula 5.3), by which:

$\Phi_{eff} = \eta_{vol} * V_s * n = \eta_{vol} * V_s * \lambda * v / \pi D_r$ (formula 5.4)

Formula 5.3 and 5.4 together:

$$V_s = \frac{\rho_a * \pi^2 * C_p * \eta_{tot} * v^2 * D_r^3}{8 * \rho_w * g * H * \eta_{vol} * \lambda}$$ (formula 5.5)

For v we take the design windspeed v_d= 3 m/s.

The other values are:

C_p= 0,27 ; η_{tot}= 0,67 ; ρ_a= 1,2 kg/m^3 ; ρ_w= 1000 kg/m^3 ;
g= 9,8 m/s^2 ; D_r= 4,5 m ; λ = 1 ; η_{vol}= 0,9

These values in formula 5 give:

$V_s = A_p * s = 24,8/H$ [lit]   (formula 5.6)

When V_s is known and the stroke length s is chosen the piston area A_p can be calculated:

$$A_p = V_s/s \qquad \text{(formula 5.7)}$$

From formula 5.2 and 5.3 it is clear that to reduce F_{peak} you must choose s as long as possible so that A_p is as small as possible. For the Diever 450 this is 250 mm.

To choose the pump cylinder it depends on what is available. In the past the WOT used iron gaspipes of 4", 5" and 6". But the iron cylinder caused trouble, because it rusts. The rust is like sandpaper for the piston. The life-time of the piston was very short (few months). Alternatives are stainless steel, bronze, brass or p.v.c. Experiences with stainless steel, bronze and brass makes clear that these materials are suitable for the use as pumpcylinder but they are difficult to obtain, especially brass cylinders. Stainless steel cylinders are often used in the food-processing industry. It is possible to cast bronze cylinders and to turn out on a big lathe. We choose p.v.c. because p.v.c. pipes are more available in developing countries. In the next table some p.v.c. pipes are chosen as pump cylinders. It gives the optimal waterlifting head calculated by formula 5.6 at s= 250 mm. It gives also the maximum admissible pressure p_{max} of the pipes.

Table 5.1: p.v.c. pipes as pump cylinders

pipe size in inch	outerxinner size in mm	optimal head in m	p_{max} in bar
8"	200x190,2	3,5	6
6"	160x150,6	5,6	7,5
5"	125x117,6	10,1	7,5
4"	110x101,6	12,2	10
3½"	90x81,4	19,0	12,5
3"	75x67,8	27,6	12,5
2"	63x53,6	44,3	16

For the 8", 6", 5" and 4" pump it is possible to take a pipe with a thinner wall, but for these pipes the admissible

waterlifting heads are smaller. In practice we found that
p.v.c. wore out when the water is sandy. Therefore it is
better to choose pipes with a thicker wall, like we did in
table 5.1.

5.5. The strength calculation of the pumps

The admissible pumprod forces depend on the admissible
pressure of the p.v.c. pipe ($\overline{F}_{pr} = p_{max} * (A_p - A_{pr})$) and on the
admissible load of the other parts of the pump like the piston
and the pumprod. Hereby we take the fatigue and the static
load. For the fatigue load I used lit. 11 and 12. The load is
a repeated load ($R = \sigma_{min}/\sigma_{max} = 0$). The safety factor is 1,2 for
the welds and 1,5 for the other parts.
The results of the calculations are put in table 5.2:

Table 5.2: strength calculations

pump in inch	rising main in inch	fatigue load $\overline{F}_{pr}$ (n=60) in kN.	static load $\overline{F}_{pr}$ (n=80) in kN.
8"	4"	10,9	18,8
6"	3"	8,3	10,8
5"	3"	7,9[*]	12,4
4"	2"	7,7[*]	11,8
3½"	2"	6,0[*]	9,0
3"	2"	4,0[*]	7,2
2"	1½"	3,0[*]	5,7

[*]These admissible values are determined by p_{max} of the p.v.c.
 pipe.

5.6. Tests of p.v.c. pumps

Two pumps with p.v.c. pump cylinders are used for the tests, a
6" pump and a 4" pump (see drawings nr. 5A and 5C). The pumps
are tested on the Diever 450 windmill and on a pump test rig.

The waterlifting head of the Diever 450 is 10,5 m, of the test rig 11,5 m (see appendix 4 and 5).

5.6.1. The aim of the tests

The aim of the tests is to determine:
1. the pumprod forces and the overshoot factor of formula 5.2.
2. the volumetric efficiency of the pump
3. the indicator diagram of the pump (force-position diagram). With this the total efficiency of the pump installation can be calculated.
4. the strength and the durability of the pump.

ad 1. The pumprod forces

The pumprod forces of both pumps are measured on the test rig, with 2" and 3" rising mains. On the Diever 450 the pumprod forces of both pumps are measured, hereby only with 2" rising mains. The peak force in the pumprod (F_{peak}) is measured for different rotation speeds. The pumprod has a force reader, which is a shaft of ϕ16 mm with strain gauges glued on both sides. An oscilloscope registered the forces on the vertical scale and the time on the horizontal scale. Hereby the peak forces and the rotation speeds could be determined.

ad 2. The volumetric efficiency

The volumetric efficiency is checked regularly by a flowmeter and a stroke counter.

$$\eta_{vol} = \frac{\text{water output per stroke}}{V_s} \times 100\% \qquad \text{(formula 5.8)}$$

ad 3. The indicator diagram

On the test rig a position reader is installed which registered the position of the piston. An oscilloscope registered the pumprod force on the vertical scale with the force reader and the position of the piston on the horizontal scale. Hereby the indicator diagram is determined. To determine the surface of the indicator diagram you get the

amount of energy input of the pump during one stroke (= E_{in}).

$$E_{in} = \frac{V_{eff} * P_{stat}}{\eta_{vol} * \eta_{mech} * \eta_{hydr}} = V_{eff} * P_{stat} / \eta_{inst} \quad \text{(formula 5.9)}$$

V_{eff} = effective water output per stroke measured by the flow
     meter and the stroke counter

P_{stat} = static pressure = $\rho_w * g * H$

With formula 5.8 and 5.9 we can calculate the total efficiency
of the pump installation η_{inst} and the product $\eta_{mech} * \eta_{hydr}$.
The hydraulic efficiency η_{hydr} separated and therefore the
mechanical efficiency of the pump η_{mech} cannot be determined
easily because the water flow and the conductor resistance in
the pump and rising mains aren't constant.

ad 4. The strength and durability

A life test of the 4" pump is done on the Diever 450 and is
still going on. The life test of the 6" pump is done on the
test rig and has finished after 4,43 million strokes.
The wear of the piston and foot valve is checked several
times.

5.6.2. The test equipment

The measuring erection of the pump test rig is given in figure
5.1. An electromotor of 3 kW, 1430 r.p.m., drive the pump by
way of a gear of three teeth belts and the Diever 450
transmission.
The crank was in the beginning variable in length like the
12PU500 but this construction wasn't strong enough. Therefore
the construction changed in a crank with a fixed cranklength,
see drawing nr. 4.
The transmission is identic to the one of the Diever 450, see
appendix 2. The drive is built on a tower of 12 meters high.
Under the tower there is a water tank of 1 meter deep in where
the pump is placed. From the pump rising mains conduct the
water upwards till a barrel which stands on the platform, 11,5
m higher than the waterlevel in the water tank. Through return
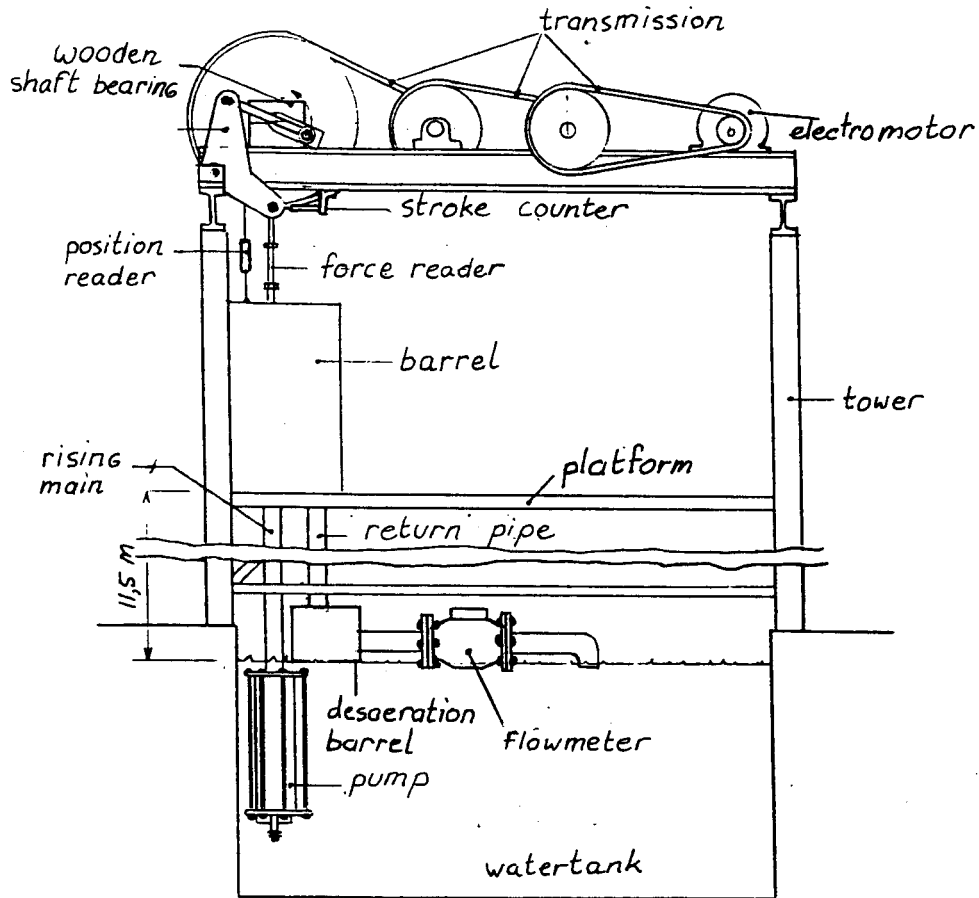pipe the water streams back in the water tank. A desaeration

figure 5.1: the pump test rig

barrel prevents that air is going with the water trough the
flowmeter. Air can cause a wrong registration of the water
flow.

The force reader is placed between the transmission and the
pumprod. The position reader is placed parallel to the pumprod
and connected between the transmission and the tower. The
stroke counter consists of an inductive reader which gives a
pulse when the transmission moves close by it and a counter
which counts the pulses.

On the Diever 450 the force reader was also coupled between
the transmission and the pumprod. A flowmeter is placed in the
return pipe. For the stroke counter we used a km-counter of a
bicycle. This is cheap but it has a very short life-time.

5.6.3. The results

1. The overshoot factor k:

the 6" pump on the test rig : 1,5 < k < 2,1

the 6" pump on the Diever 450: 1,3 < k < 1,7

the 4" pump on the test rig : 1,5 < k < 2,0

the 4" pump on the Diever 450: 1,4 < k < 2,0

Remarks:

A higher valve lift caused a bigger F_{peak} with 2" rising mains and didn't show any obvious difference of F_{peak} with 3" rising mains. The average k-value on the Diever 450 is lower than on the test rig.

2. The volumetric efficiency:

the 6" pump: 82% < $\cap_{vol}$ < 107%

the 4" pump: 92% < $\cap_{vol}$ < 99%

Remarks:

The low value of 82% was caused by a leaking foot valve. The high value of 107% (more than 100%!) was caused by the inertia-effect: the kinetic energy of the accelerated water column is converted into an extra delivery.

A higher rotation speed gave a higher $\cap_{vol}$.

3. The indicator diagram:

the 6" pump: 62% < $\cap_{inst}$ < 83%

the 4" pump: 53% < $\cap_{inst}$ < 56%

Remarks:

A lower $\cap_{vol}$ caused by a leaking foot valve gave a higher $\cap_{inst}$.

A higher valve lift of the foot valve only caused a higher $\cap_{inst}$ for the 6" pump but not for the 4" pump.

4. The life tests:

The 6" pump on the test rig: The wear of the wall thickness of the p.v.c cylinder was 0,2 till 0,5 mm after 3,37 million strokes. Therefore it is wise to choose a p.v.c.pipe with a thick wall. The wear of the piston cup was negligible. The volumetric efficiency of the pump varied during the test from 82% till 107% with an average value of 98%.

The 4" pump on the Diever 450: The pump is placed on the 24th
of may in 1988 and has made more or less 2,5 million strokes
till now (may 1990). Nothing has broken yet. The wear of the
lip thickness of the leather piston cup is about 0,2-2 mm.

5.7. The choice of the p.v.c. pipe as pump cylinder

In par. 5.5 we found the admissible loads for different pumps
with p.v.c. cylinders and in par. 5.4 the optimal waterlifting
head with R_t= 120 mm. In par. 5.6 we found the overshoot factor
k for the 4"pump (with R_t= 120 mm) between 1,4 and 2,0 and for
the 6"pump (with R_t= 60 mm) between 1,3 and 2,1. These values
are a little bit lower than the values of the C.W.D. (lit.10)
because of the four bars mechanism. Another point is the big
fluctuation in the k-values. That's why it isn't possible to
predict the pumpforces accurately. For safety we choose the
value: k= 2,0 for R_t= 120 mm. Now we can calculate the maximum
admissible waterlifting heads (H_{max}) of the pumps with formula
5.2:

Table 5.3: maximum admissible waterlifting heads

pump in inch	Outerxinner diameter in mm	risingmain in inch	pumprod in inch	H_{max} in m
8"	200x190,2	4"	$^3/_4$"	7,5
6"	160x150,6	3"	½"	7,5
5"	125x117,6	3"	½"	18,4
4"	110x101,6	2"	½"	16,9
3½"	90x81,4	2"	½"	27,9
3"	75x67,8	2"	½"	34,2
2"	63x53,6	1½"	½"	50

Table 5.4: Recommendations for choosing the pumpcylinder

waterlifting head	pump diameter in inch	rising main in inch
from 0 till 5 m	8"	4"
from 5 till 7 m	6"	3"
from 7 till 12 m	5"	3"
from 12 till 16 m	4"	2"
from 16 till 25 m	3½"	2"
from 25 till 34 m	3"	2"
from 34 till 50 m	2"	1½"

The design windspeed as functions of the waterlifting head for table 5.4 can be calculated with formula 5.5 and the data in par. 5.4:

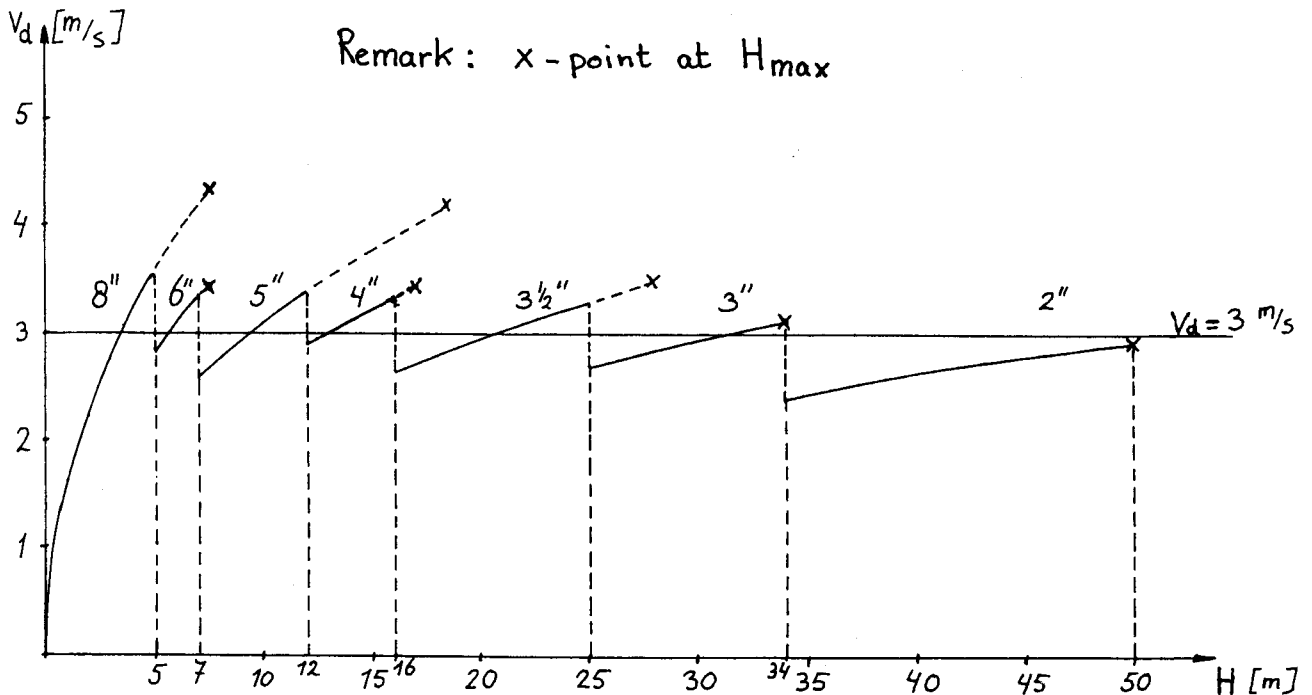$$v_d \approx 9,51 \ * \ \sqrt{\{(A_p - A_{pr}) * H\}}$$

figure 5.2: the design windspeed as functions of the waterlifting head

CHAPTER 6. THE TOWER

6.1. General

The evaluation of the 12PU500 in par. 1.2 has learned us that
the tower of 6 m high is too low and that the tower is
difficult to transport. For the Diever 450 we chose a tower
with a height of 10 meters. The tower is a frame consisting of
angle irons and is welded in 10 parts which are connected
together with bolts. The welded parts are transportable all
together on a roofrack of a landrover.

The construction has four legs which are made of ∟50x50x5, the
horizontals and diagonals of ∟30x30x3. The towerpipe is made
of 4" gaspipe (see drawings nr. 6 and 6A). At one side there
are stairs till a height of 6 meters (drawing nr. 6B). It
continues at two opposite sides of the tower till 9 meters. We
chose for the last 3 meters two stairs at opposite sides to
stay away from the turning rotor. From the stairs you can step
on the seat which is connected to the head (see drawing nr.
6C). On this seat it is easy and safe to lubrificate the
shaftbearings and the bearings of the transmission or to do
other maintenance work. This is safer then standing on a
platform which is connected to the tower.

Aerodynamic forces, gravity forces and pumprod forces act on
the tower. These are calculated for two situations: for
maximum operating conditions and for storm conditions.
In the calculations the following conditions are assumed:
- Survival windspeed of 40 m/s.
- At the windspeed of 12 m/s it is possible that the rotor
  temporarily faces the wind perpendicular.
- The influence of rain, snow and glazed frost is neglected.
- The stresses in the legs will be maximal when the wind
  attacks the tower in a diagonal way. This is explained in
  fig. 6.1.
Assuming the tower to hinge on foundation blocks B and D
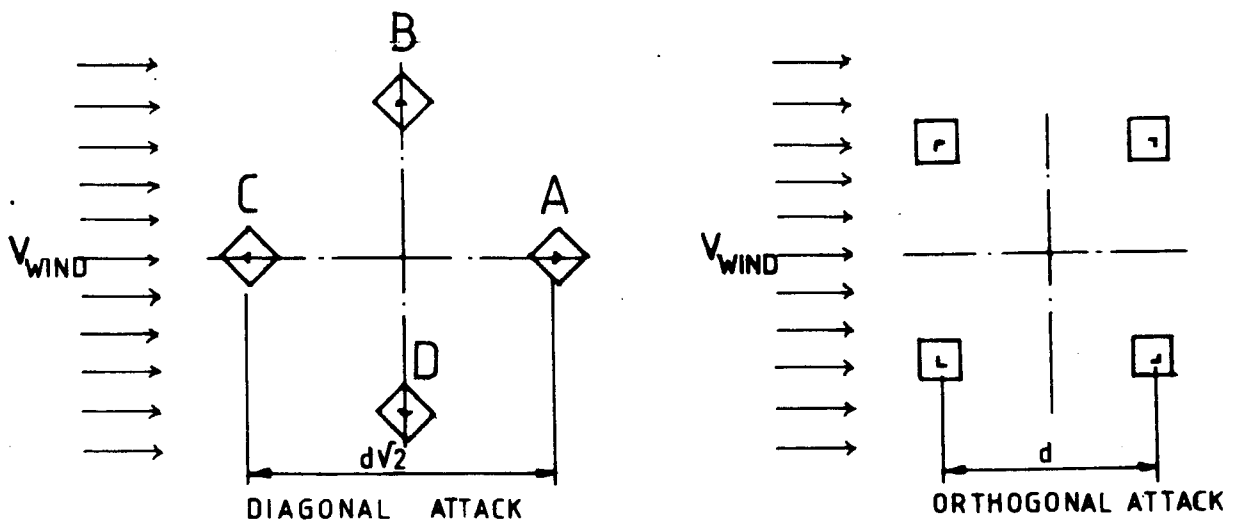during diagonal attack, the total wind force is divided among

figure 6.1: top view on the tower

legs A and C, loading A to buckle and C to tensile forces.
The vertical load is devided over all four legs in both
situations. But the horizontal load (wind load) and the
moments on the tower top cause a barforce in leg A
(compression) and C (tensile) under diagonal attack which is
$\sqrt{2}$ times higher than in leg C and D (compression) and A and B
(tensile) under orthogonal attack. Also the projected area of
the tower under diagonal attack is bigger than under
orthogonal attack. Therefore in the following calculations
only the diagonal attack of the wind on the tower is taken
into account.

Calculations on the tower are divided in the following parts:

Forces on the tower top (6.2)

1. at operating conditions (v= 12 m/s)

2. at storm conditions (v= 40 m/s)

Forces on the tower construction (6.3)

1. at operating conditions

2. at storm conditions

Stresses in the towerlegs (6.4)

1. buckling

2. tensile

Foundation (6.5)

Forces on the tower during hoisting (6.6)

6.2 Forces on the tower top

6.2.1. At operating conditions (v= 12 m/s)

If we have seen in chapter 3 at v= 12 m/s the rotor turns out the wind. But because of fluctuating windspeed and wind direction it is possible that the rotor temporarily faces the wind perpendicular at v= 12 m/s. Therefore the next conditions are assumed:

- v= 12 m/s
- the rotor is turning at full speed (n= 80 r.p.m.)
- the main vane is parallel to the wind direction
- Gyroscopic effects of the rotor cause a moment on the tower

Data:		Value:	Unit:
v	windspeed	12	m/s
C_T	rotor trust coefficient	0,89	--
C_D	drag force coefficient	1,2	--
d_{rx}	distance c.o.g. (= center of gravity) rotor to y-axis	0,8	m
d_{rz}	distance c.o.g. rotor to tower top	0,48	m
d_{vx}	distance c.o.g. main vane to y-axis	2,7	m
d_{ax}	distance c.o.g. side vane to y-axis	0,37	m
d_{ay}	distance c.o.g. side vane to x-axis	2	m
d_{hx}	distance c.o.g. head to y-axis	0	
d_{hy}	distance c.o.g. head to x-axis	0	
$F_{pr\ max}$	maximum pumprod force, see below	11200	N
m_r	mass rotor	206	kg
m_h	mass head	50	kg
m_v	mass main vane	56	kg
m_a	mass side vane	13	kg
m_{tr}	mass transmission	8	kg
M_{gyr}	maximum gyroscopic moment of the rotor, see below	1540	Nm
R_r	rotor radius	2,25	m
ρ_a	air density	1,2	kg/m^3

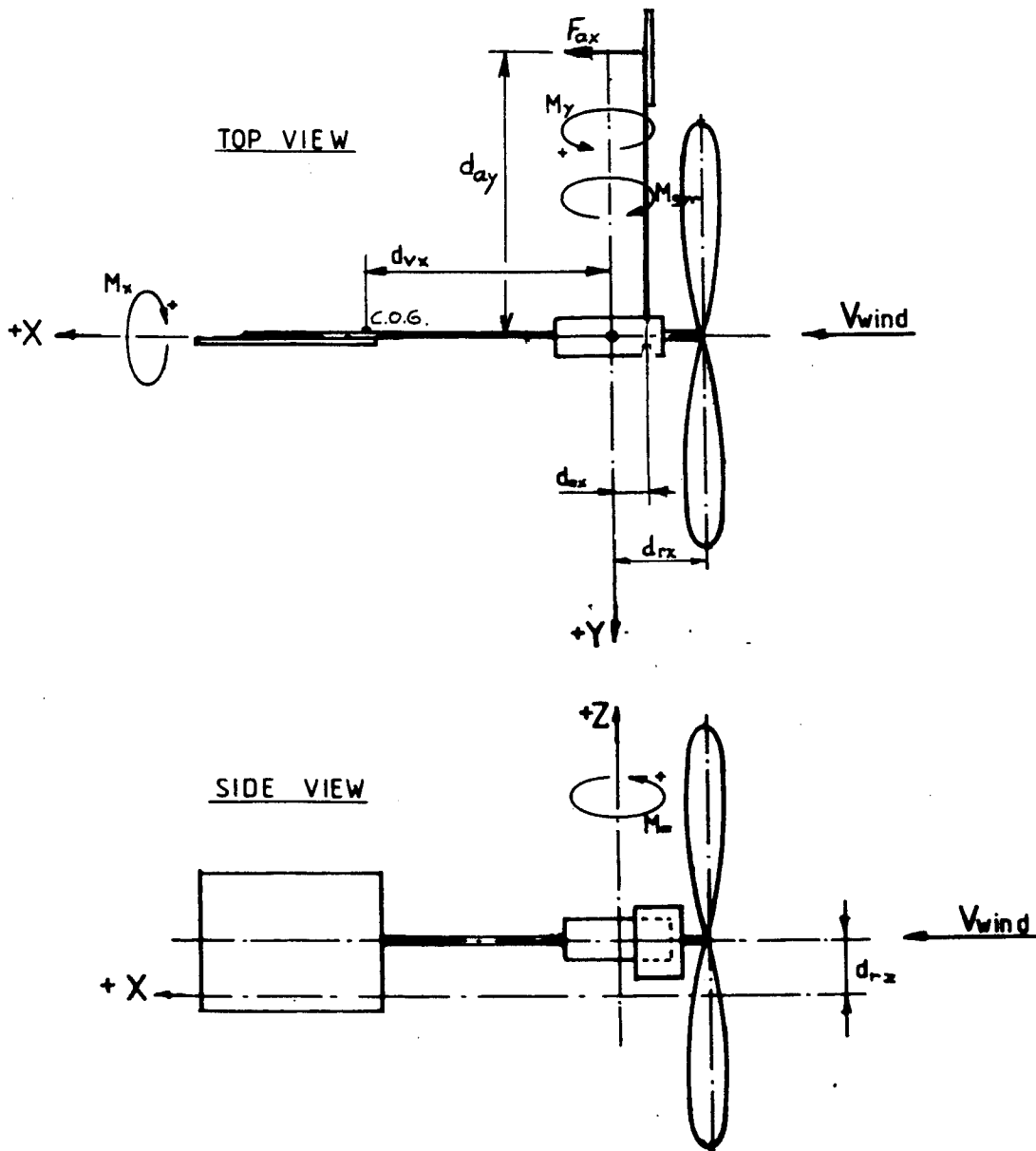To calculate $F_{pr\ max}$ we use the table in par. 5.7. When we take

figure 6.2: situation in operating conditions

for k= 2,0 and H the upper limit of the chosen interval for each pump size we can calculate the pumprod force with formula 5.2. The 4"pump with H= 16 m has the highest force. Therefore we take this pump and waterlifting head for the calculations here.

The gyroscopic moment $M_{gyr}= J*\Omega*\Omega_{yaw}$:

J= moment of inertia of the rotor= 291 kgm²

Ω= angular speed of the rotor= 80/60*2π= 8.4 rad/s

Ω_{yaw}= angular speed of the head= maximum of 0.63 rad/s (this is
    a value measured by the CWD on the CWD5000 which is
    comparable to the Diever 450)

The forces and moments on the tower top:

$F_X = C_T * \frac{1}{2} * \rho_a * v^2 * \pi * R^2 + F_{ax} =$ 1267 N

$F_{ax} = C_D * \frac{1}{2} * \rho_a * v^2 * A_a =$ 44 N

$F_Z = -(m_h + m_{tr} + m_r + m_v + m_a) * g - F_{pr\ max} =$ -14463 N

$M_X = m_a * d_{ay} * g$ 255 Nm

$M_Y = (m_v * d_{vx} - m_a * d_{ax} - m_r * d_{rx}) * g + (F_X - F_{ax}) * d_{rz} +$
$\qquad M_{gyr} =$ 1947 Nm

$M_Z =$ friction in the yaw bearing (is neglected) 0 Nm

6.2.2. At storm conditions (v= 40 m/s)

In storm conditions it is assumed that;
- the rotor is not turning
- the rotorplane and both vanes are parallel to the wind
  direction

Data:		Value:	Unit:
v	windspeed	40	m/s
A_{hs}	side area head	0,28	m²
A_{rs}	side area rotor	1,2	m²
d_{ry}	distance c.o.g. rotor to x-axis	0,8	m
d_{vx}	distance c.o.g. main vane to y-axis	2,46	m
d_{vy}	distance c.o.g. main vane to x-axis	0,24	m
d_{vz}	distance c.o.g. main vane to x-axis	0,43	m
d_{ax}	distance c.o.g. side vane to y-axis	2	m
d_{ay}	distance c.o.g. side vane to x-axis	0,37	m
d_{avx}	distance aerodynamic center to y-axis	3,4	m
m_{p+pr}	mass of piston + pumprod	37	kg

Calculations and results:

$F_{h,s} =$ sideways force on head=
$\qquad C_D * \frac{1}{2} * \rho_a * v^2 * A_{hs}$ 322 N

$F_{r,s} =$ sideways force on rotor=
$\qquad C_D * \frac{1}{2} * \rho_a * v^2 * A_{rs}$ 1382 N

As the sum of moments around z-axis = 0 we find;

$F_{v,y} * d_{avx} - F_{r,s} * d_{ry} = 0$
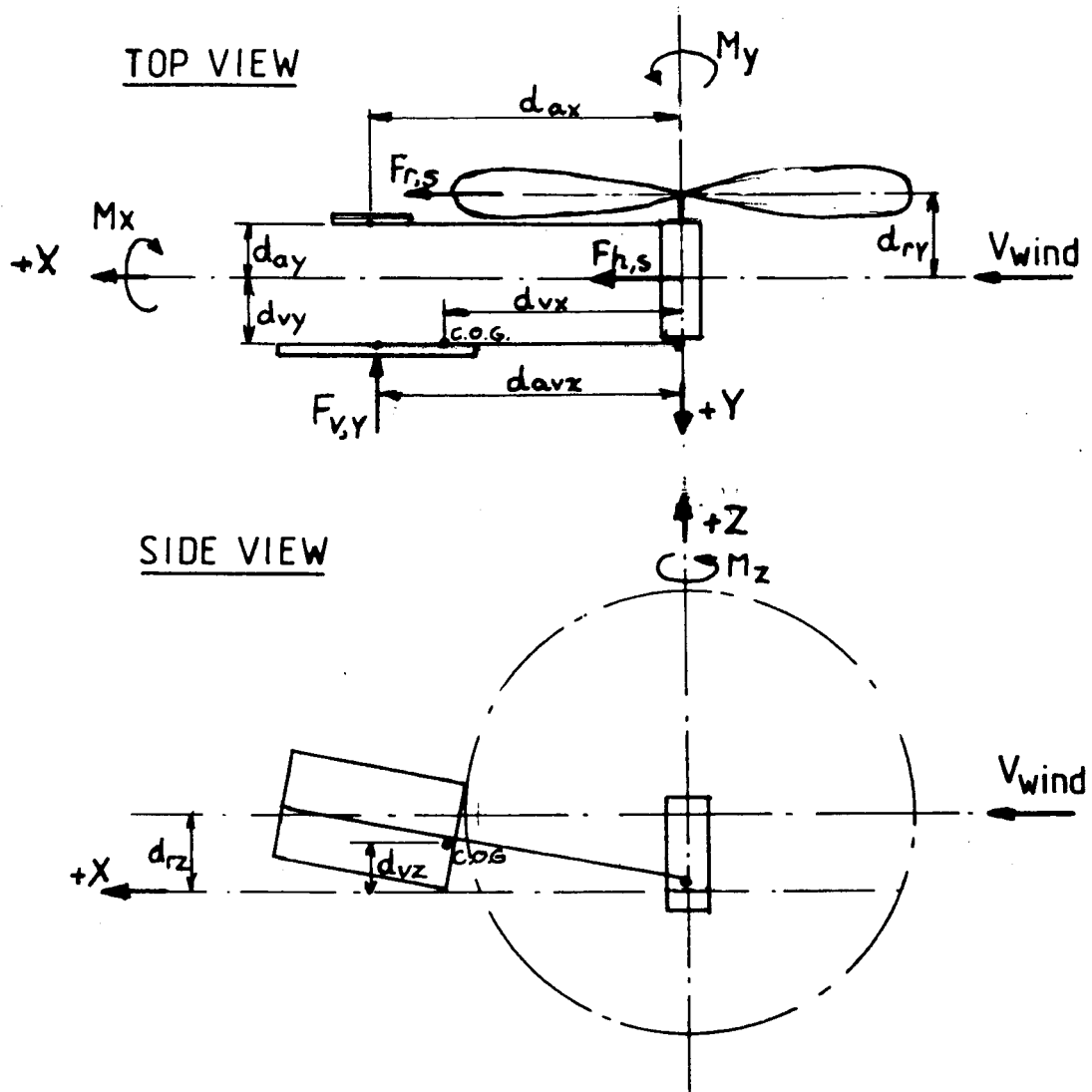
TOP VIEW

SIDE VIEW

figure 6.3: situation in storm conditions

$F_{v,y}$ force on main vane in y-direction   325   N

The forces and moments on the tower top:

$F_x = F_{h,s} + F_{r,s} =$ 1704   N

$F_y = -F_{v,y} =$ -325   N

$F_z = -(m_h + m_{tr} + m_r + m_v + m_a + m_{p+pr})*g =$ -3626   N

$M_x = (-m_v*d_{vy} + m_r*d_{ry} + m_a*d_{ay})*g + F_{v,y}*d_{v2} =$ 1724   Nm

$M_y = (m_v*d_{vx} + m_a*d_{ax})*g + F_{r,s}*d_{rz} =$ 1811   Nm

$M_z =$ friction in the yaw bearing (is neglected)   0   Nm

6.3. Forces on the tower construction

First the load is calculated for the operating and storm
conditions and the reaction forces in the supporting points.
With these results the barforces in the legs can be calculated
as followed:
The barforce of a leg is equal to the vertical reaction force
in the supporting point of the leg divided by the cosines of
the angle between the leg and the vertical, for example:
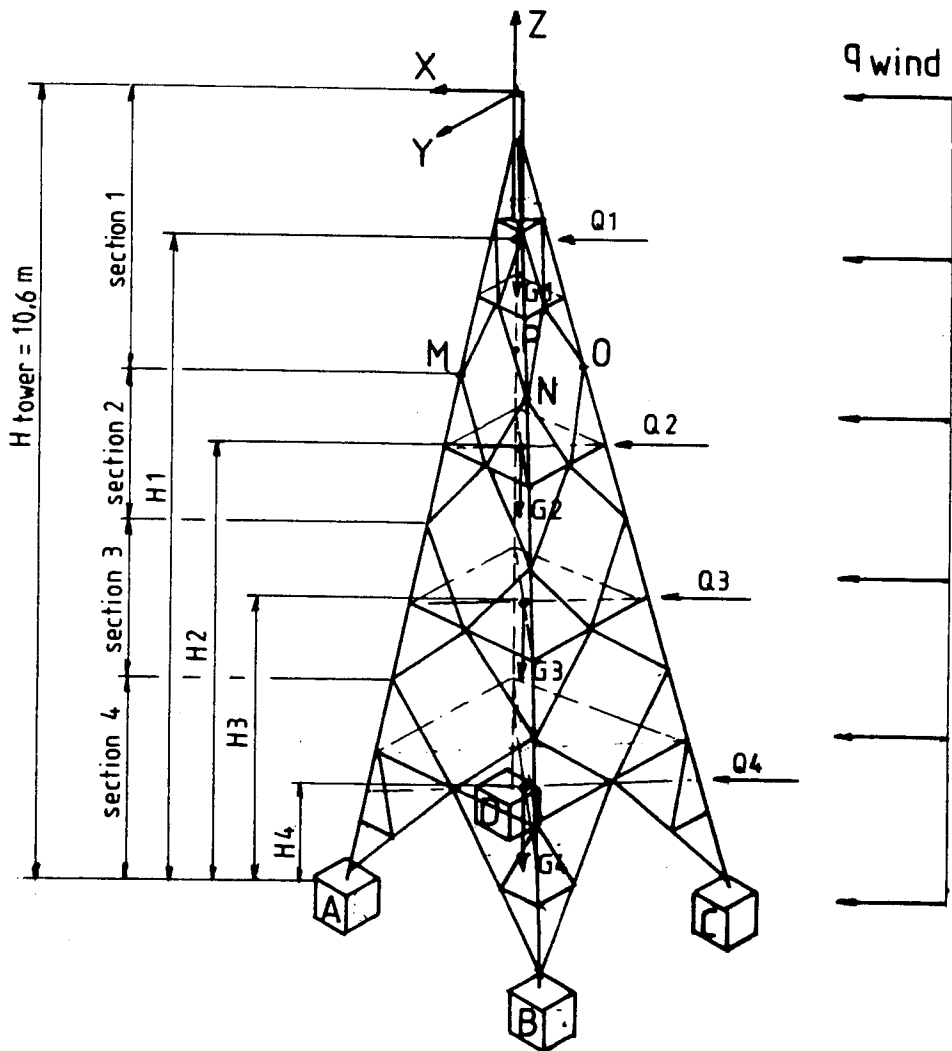
$F_{leg\ AE} = F_{VA}/\cos 12°$.

figure 6.4: diagonal attack of the wind on the tower
construction

The following simplifications are made:

- the tower is split up in four sections, see fig. 6.4
- all the legs are exposed to the wind (no wind shadow)
- the rising main ends 2,5 m below the tower top
- the projected length of the horizontals is the real length
  (l_h) multiplied by $\frac{1}{2}\sqrt{2}$, the projected area of the horizontals
  is $\frac{1}{2}\sqrt{2}*l_h*t$ (t= width of the profile)
- the total projected length of the diagonals (l_d) is equal to
  2*4*height of the section, the projected area of the
  diagonals is the length multiplied by $t*\frac{1}{2}*(\sqrt{2} + \frac{1}{2}\sqrt{2})=$
  1,06*t
- the point of impacts (=p.o.i.)of the section is in the
  middle

6.3.1. At operating conditions (v= 12 m/s)

Forces and stresses an the tower are caused by:
- forces and moments on the tower top, see par. 6.2.1
- the wind load on the tower construction
- the tower weight
- the weight of the pump and rising main

Data:		Value:	Unit:
A_{proj1}	projected area section 1	1,64	m²
A_{proj2}	projected area section 2	1,20	m²
A_{proj3}	projected area section 3	1,26	m²
A_{proj4}	projected area section 4	2,24	m²
C_D	drag coefficient: for angle irons	1,6	--
	for rising main and towerpipe	1,2	--
H_1	height p.o.i. of section 1	8,5	m
H_2	height p.o.i. of section 2	6	m
H_3	height p.o.i. of section 3	4	m
H_4	height p.o.i. of section 4	1,5	m
m_1	mass of section 1	107	kg
m_2	mass of section 2	67	kg
m_3	mass of section 3	69	kg
m_4	mass of section 4	119	kg
m_t	mass of tower	362	kg

m_{pu+rm} mass pump + rising main   140   kg

Forces on the tower:

The tower experiences the forces and moments on the tower top calculated in par. 6.2.1, and:

		Value:	Unit:
Q_1 =	wind load on section 1=		
	$C_D * \frac{1}{2} * \rho_a * v^2 * A_1$	227	N
Q_2	wind load on section 2	166	N
Q_3	wind load on section 3	174	N
Q_4	wind load on section 4	310	N
G_1 =	weight of section 1= $-(m_1 + m_{pu+rm})*g$	-2421	N
G_2 =	weight of section 2= $-m_2 *g$	-660	N
G_3 =	weight of section 3	-680	N
G_4 =	weight of section 4	-1170	N

Calculations and results:

First the reaction forces in the supporting points M, N, O and P of section 1 are calculated and then in A, B, C and D. Only the vertical reaction forces are calculated because these are necessary for the calculations of the barforces in the legs.

In M, N, O and P:

F_{vM}= vertical reaction force in M=

$\frac{1}{4}*(-F_z + G_1) + (M_y + F_x*3,6 + Q_1*1,8)/1,36$= 9307 N

F_{vN}= vertical reaction force in N=

$\frac{1}{4}*(-F_z + G_1) - (M_x - F_y*3,6)/1,36$= 4033 N

F_{vO}= vertical reaction force in O=

$\frac{1}{4}*(-F_z + G_1) - (M_y + F_x*3,6 + Q_1*1,8)/1,36$= -865 N

F_{vP}= vertical reaction force in P=

$\frac{1}{4}*(-F_z + G_1) + (M_x - F_y*3,6)/1,36$=

4409 N

In A, B, C and D:

F_{vA}= vertical reaction force in A=

$\frac{1}{4}*(-F_z + G_1 + G_2 + G_3 + G_4) + (M_y + F_x*10,6 + Q_1*8,5 + Q_2*6 + Q_3*4 + Q_4*1,5)/4,2$= 4849 + 4634= 9483 N

F_{vB}= vertical reaction force in B=
¼*(-F_z + G_1 + G_2 + G_3 + G_4) - (M_x -
F_y*10,6)/4,2= 4849 - 61=
4788 N

F_{vC}= vertical reaction force in C=
4849 - 4634= 215 N

F_{vD}= vertical reaction force in D=
4849 + 61= 4910 N

Conclusion:

The maximum compression and
tensile loads occur respectively
on legs AE and OS. These loads
will be compared with loads under
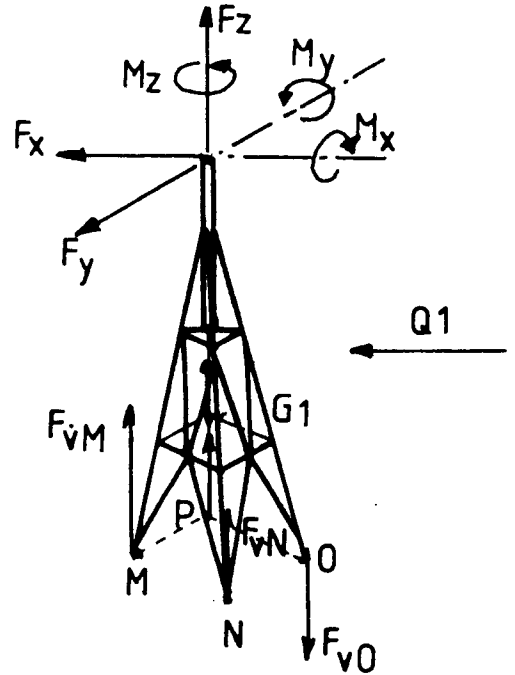storm conditions, see next
paragraph.

figure 6.5: load and
reaction forces on section
1

6.3.2. At storm conditions (v= 40 m/s)

Forces and stresses on the tower are caused by:
- forces and moments on the tower top, see par 6.2.2.
- the wind load on the tower
- the tower weight
- the weight of pump and rising main, both filled with water

Data:		Value:	Unit:
v	windspeed	40	m/s
m_w	mass of the water column in pump and rising main	31	kg

Forces on the tower:

The tower experiences the forces and moments on the tower top
and:

Q_1	wind load on section 1	2520	N
Q_2	wind load on section 2	1850	N
Q_3	wind load on section 3	1940	N
Q_4	wind load on section 4	3450	N

G_1	weight of section 1= $-(m_1 + m_{pu+rn} + m_v)*g =$	-2724	N
G_2	weight of section 2	-660	N
G_3	weight of section 3	-680	N
G_4	weight of section 4	-1170	N

Calculations and results:

The vertical reaction forces in M, N, O and P:

F_{vM}= 1588 + 9178= 10766 N

F_{vN}= 1588 - 2128= -540 N

F_{vO}= 1588 - 9178= -7590 N

F_{vP}= 1588 + 2128= 3716 N

In A, B, C and D:

F_{vA}= 2215 + 15554= 17769 N

F_{vB}= 2215 - 1231 = 984 N

F_{vC}= 2215 - 15554= -13339 N

F_{vD}= 2215 + 1231 = 3446 N

Conclusions:

Maximum loads occur on legs AE (compression) and CG (tensile).
These loads are used for the stress calculations. The storm
conditions are decisive for the tower construction and also
for the foundation block dimensions.

6.4. Stresses in the towerlegs

As already given in par. 6.2 the barforces in the legs are:

F_{bar}= $F_{reaction}$/cos 12°

This means that the maximum compression force is:

F_{bar}= 17769/0,955= 18606 N

The maximum tensile force is: F_{bar}= 13339/0,955= 13967 N

The compression force is decisive for the strength of the legs
because of buckling. Therefore the maximum buckling stress is
calculated:

Data:		Value:	Unit:
l_b	buckling length	1000	mm

i_{min}	minimum radius of gyration L50x50x5	9,7	mm
A	cross-section area L50x50x5	480	mm²
F_{bar}	barforce (maximum compression)	18606	N
A_{bolt}	stress area of M10 bolt	74	mm²

The connections of the legs consists of three bolts M10

Calculations and results:

λ	slenderness ratio= l_b/i_{min}	103	--
ω	buckling coefficient (see table in lit. 13)	2,12	--
σ_{buckle}	buckling stress= $\omega * F_{bar}/A$	82	N/mm²
τ_{bolt}	shear stress in bolt M10= $F_{bar}/(3*A_{bolt})=$	83,5	N/mm²

Conclusion:

As the maximum admissible stress is 160 N/mm² and the
calculated stress is 82 N/mm² it is obvious that the legs are
strong enough.
The admissible shear stress for bolts is: $\bar{\tau}$= 0,7*160= 112
N/mm². The calculated shear stress is smaller.

6.5. The foundation

The calculated maximum reaction force in point C is decisive
for the dimensions of the foundation blocks. The blocks must
be heavy enough to overcome the tensile forces.

Data:		Value:	Unit:
H_B	height of block	0,9	m
W_B	width of block	0,9	m
L_B	length of block	0,9	m
ρ_c	density of concrete	2100	kg/m³
F_{VC}	vertical reaction force in C	13327	N

Calculations and results:

G_B	block weight= $H_B * W_B * L_B * \rho_c * g$	15003	N

Conclusion:

The block is heavy enough to take the maximum tensile force

6.6. Forces on the tower during hoisting

The installation of the windpump can be done by two methods.
The first method is the method of the 12PU500, explained in
appendix 6 and lit. 3. Only the tower of the Diever 450 is
higher. Therefore the poles must be higher (about 6 m).
The other method is as followed: First the whole windpump is
assembled in the horizontal position. two legs are connected
to the foundation with each a bolt M16, see drawing nr. 6E.
This bolt is the hinge when the windmill is hoisted
vertically. The load during hoisting is caused by the weight
of the tower and head with rotor, transmission and vanes.
Windpressure is not taken in account. The tensile force in the
steel cable causes compression forces in the tower. It is
assumed that the cable support yoke has a height of 5,8 m (see
fig. 6.6 and drawing nr. 7).
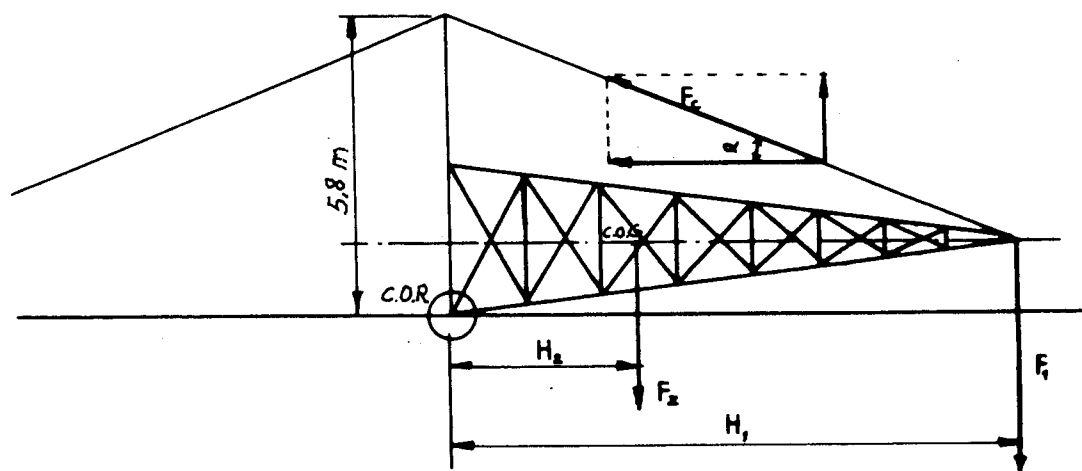
figure 6.6: situation during hoisting

The stresses in the legs and near the centre of rotation
(c.o.r.) are checked:

Data:		Value:	Unit:
H_2	horizontal distance between c.o.r. and c.o.g. of the tower	4,9	m
H_1	height tower till head, see fig. 6.6	10	m
m_t	mass of tower	362	kg
m_{htot}	mass of head, rotor, transmission and		

	vanes	313	kg
α	angle between horizontal and cable	21	degrees
W_b	bending moment of resistance L50x50x5	3050	mm^3

Calculations and results:

F_c	tensile force in cable=		
	$(H_2*m_t + H_1*m_{htot})*g/(H_1*\sin \alpha)$	12166	N
F_{bar}	bar force in tower leg= $(F_c*\cos \alpha)/2$	5134	N
σ_{buckle}	buckling stress= $F_{bar}*\omega/A$	22,7	N/mm^2
$F_{c.o.r.}$	reaction force in c.o.r.	5213	N
D_{min}	minimum diameter bolts in c.o.r.=		
	$4/\pi * \sqrt{(F_{c.o.r.}/160)}$	7,3	mm

Conclusions:

- The towerlegs are strong enough to take the barforces which appears during hoisting.
- The M16 bolts in c.o.r. are strong enough too.
- The tensile force in the cable is 12166 N. This is too high to pull by manpower even with pulleys. It is necessary to use a linch.
- Calculating with a calculation factor of 1,5 and a safety factor of 6 the minimum breaking load of the cable must be 109 kN.

CHAPTER 7. PROPERTIES AND CHARACTERISTICS

7.1. The input-output relation

For installing a windpump at a given site, the next parameters
of the site are important:
- the average (annual) windspeed ($\bar{v}$)
- the waterlifting head (H)

The power input at the average windspeed $\bar{v}$ is:

$P_{wind} = \frac{1}{2} {}^{*}\rho_a {}^{*}A_r {}^{*}\bar{v}^3$

The load must be related to the input power. Therefor we have
to choose the design windspeed (v_d). From lit. 10 and 14 it is
recommended to choose the design windspeed more or less equal
to the average windspeed:

$v_d \approx \bar{v}$

By using formula 5.5 we can determine the optimal swept volume
of the pump. A higher design windspeed results in a bigger
pump, which causes higher pumprod forces. However the range of
the design windspeed is limited by the admissible pumprod
forces, see chapter 5. But we must not forget that in chapter
2 we chose $v_d = 3$ m/s for the rotor. That means that the rotor
design is optimal only at the windspeed v= 3 m/s.

The maximum power coefficient of
the Diever 450 isn't measured in
a windtunnel yet. We assume that
the calculation of the C_p-value
of par. 2.4 gives the correct
value: $C_{p,max} = 0,27$.
Fig. 7.1 gives the C_p-value
related to the tipspeed ratio,
see lit. 16. Off course this
graph must be verified by a
windtunnel test.
The power of the rotor is:

$P_r = C_p {}^{*}P_{wind}$
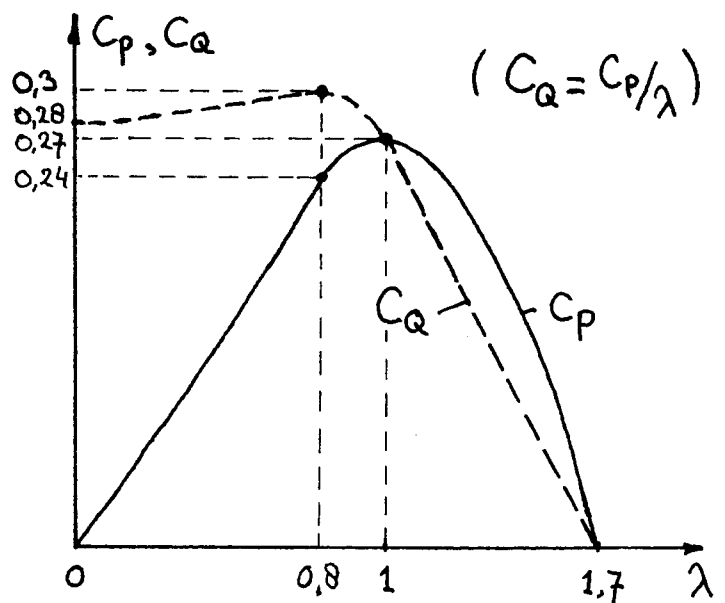
figure 7.1: C_p and C_Q
as function of λ

There are other losses of power beside of the rotor:
- in the transmission by friction in the bearings. The
  efficiency of the transmission with wooden bearings (η_{tr}) is
  between 0,92 and 0,98.
- in the pump installation. The efficiency of the pump
  installation (η_{inst}) consists of:
  * mechanical efficiency (η_{mech})
  * hydraulic efficiency (η_{hydr})
  * volumetric efficiency (η_{vol})
  The measurements in chapter 5 shows that efficiency of the
  pump installation η_{inst} varied between 0,53 and 0,83. The
  volumetric efficiency η_{vol} varied between 0,82 and 1,07.

The total efficiency of the pump installation and the
transmission is: $\eta_{tot} = \eta_{tr} * \eta_{inst}$.
We assume that: $\eta_{inst} = 0,70$ $\left.\begin{array}{c} \\ \\ \end{array}\right\}$ $\longrightarrow$ $\eta_{tot} = 0,67$
$\eta_{tr} = 0,95$

The relation between the input- and output power is:
$P_{out} = C_p * \eta_{tot} * P_{wind}$ (formula 7.1)
(The product $C_p * \eta_{tot}$ is called the overall efficiency of the
windpump)
The output power is also equal to:
$P_{out} = \rho_w * g * H * \Phi_{eff}$ (formula 7.2)

with: $\Phi_{eff} = \eta_{vol} * V_s * n$ (formula 7.3)

The formulas 7.2 and 7.3 together:
$P_{out} = \rho_w * g * H * \eta_{vol} * V_s * n$ (formula 7.4)

The volumetric efficiency can variate from 0,9 for low
rotation speeds till 1,1 for high rotation speeds. We assume
that η_{vol} is constant, namely 0,9.

The relation between the windspeed and the rotation speed of
the rotor is put in a graph, see fig. 7.2. The graph is made
by using the graphs in fig. 3.7 and 3.8 and the λ-values in
fig. 2.6. The λ-values are measured with a 4" pump and a

waterlifting head of 10,5 m, see par. 2.6.

The rotation speed in formula:

n= v*cos δ * λ/(π*D$_r$)   (formula 7.5)
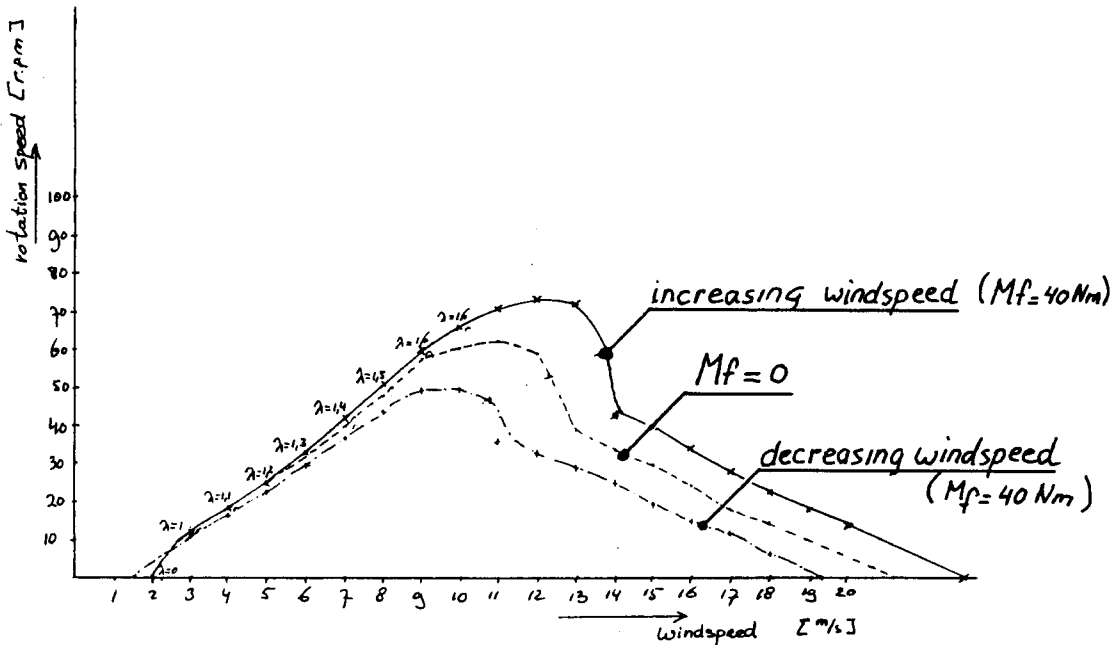

figure 7.2: the rotation speed as function of the windspeed

Now we can make the graph of the power output as function of the windspeed. When we use the formula 5.6 of par. 5.4 (with v_d= 3 m/s): V_s= 24,8*10^{-3}/H [m^3] in formula 7.4 then the output power is:

P_{out}= 243*n   [W], n in [r.p.s.]   (formula 7.6)
    (= 4,05*n [W], n in [r.p.m.])

In fig. 7.3 the power output is given as function of the windspeed, without the influence of the friction in the yaw bearing.

The formula 7.6 is only valid for the design windspeed of 3 m/s. If the average windspeed isn't equal to 3 m/s then it is more efficient to choose another swept volume of the pump. For example: $\bar{v}$= 4 m/s

    v_d= $\bar{v}$= 4 m/s

Assuming that λ_d=1 and $C_{p,max}$= 0,27 the optimal swept volume is:

V_s= 40*10^{-3}/H   [m^3] → P_{out}= 392*n
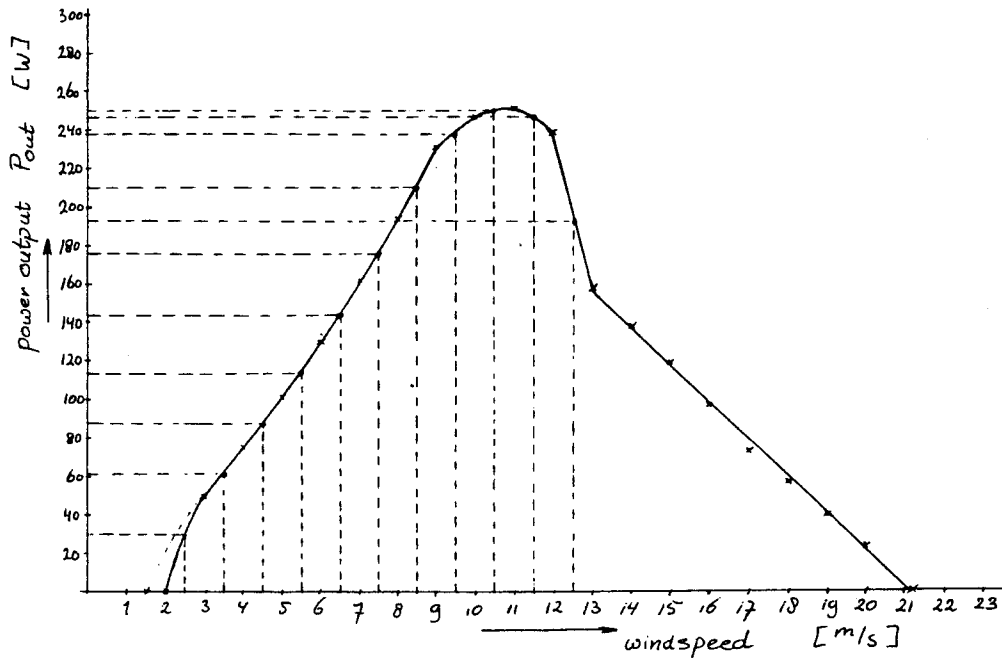
figure 7.3: the power output as function of the windspeed

7.2. Prediction of the output

The calculation of the average power output of a windpump can be made by a simple equation of formula 7.1:

$$\bar{P}_{out} = C_p * \eta_{tot} * \tfrac{1}{2} * \rho_a * A_r * \bar{v}^3 \qquad \text{(formula 7.7)}$$

This formula is simple but not very accurate. A more accurate method is using a frequency distribution of the windspeeds, see lit. 1 and 5. In the following example the graphs of fig. 7.3 and 7.4 are used. Fig. 7.4 gives an example of the frequency distribution of hourly windspeeds in Babatpur, India of the month March over the years 1976 and 1979. If such a frequency distribution is known the next method can be used for a more accurate calculation of the output:

- First we take $\bar{v}_h$ as the hourly average windspeed of an interval, which is the windspeed in the middle of the interval, see fig. 7.4.
- From fig. 7.3 we find P_{out} at the windspeed v_h:
  $\bar{P}_{out}(\bar{v}_h)$= average power output of an interval
- The water output: $\Phi_{eff}= P_{out}/(\rho_w * g * H)$ [m³/s]
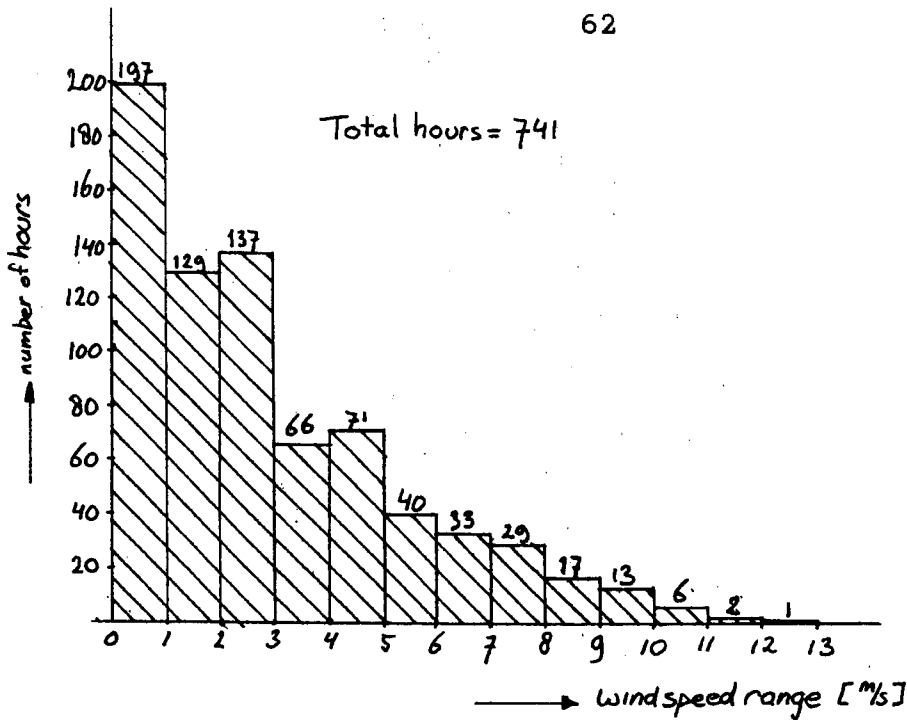 $\qquad\qquad\qquad = 3600 * P_{out}/(\rho_w * g * H)$ [m³/h]

figure 7.4: frequency distribution of hourly windspeeds in Babatpur

the total water output in the month=

$$\Phi_{month} = 3600/(\rho_w * g * H) \; * \; \sum_{v_{start}}^{v_{max}} t * \bar{P}_{out}(\bar{v}_h) \; [m^3/month]$$

with:

v_{max}= maximum windspeed by which the rotor turns (in the example v_{max}= 21 m/s)

v_{start}= starting windspeed (in the example 2 m/s)

t= number of hours per interval, see fig. 7.4

H= waterlifting head (in the example 10,5 m)

Φ_{month}= 144 + 141 + 219 + 160 + 166 + 179 + 108 + 52 + 17 + 7=
= 1318 m³/month (= 42,5 m³/day)

Remark: the rotor stops turning at v_{stop} which is lower than v_{start}. So sometimes there can be output when the windspeed is between v_{stop} and v_{start}. In the calculation we neglect it because we don't know how many times and the output is very small in comparison to the total output.

The calculation of the output with formula 7.7:
the average windspeed $\bar{v}$= 3,2 m/s

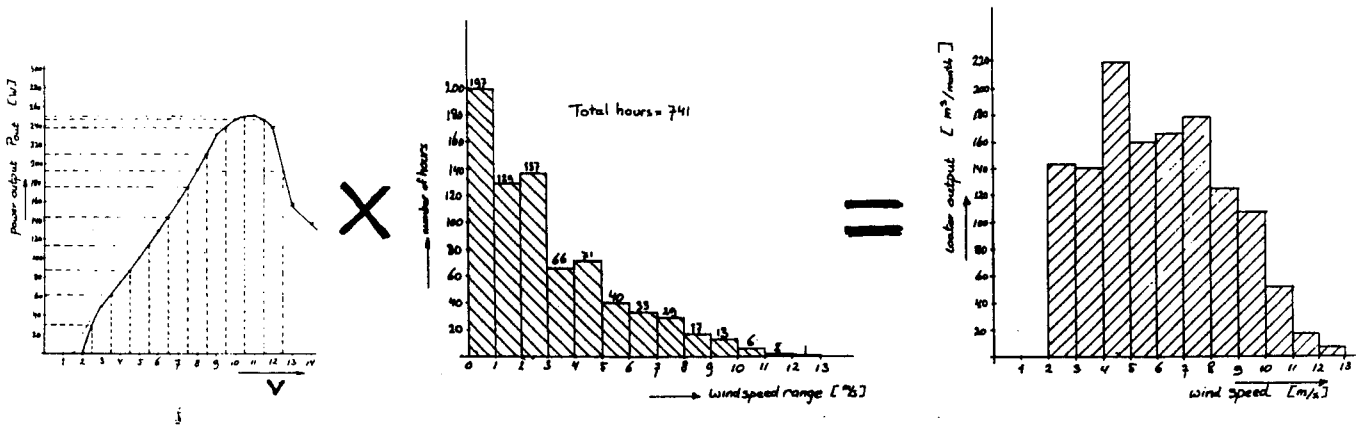figure 7.5: the calculation of the water output

$\overline{P}_{out}$= 60 W

Φ_{eff}= 50 m³/day

Φ_{month}= 1562 m³/month

This is almost 20% more than the first result. The formula 7.7 is less reliable but it is a way to predict the water output roughly if only the average windspeed is known.

7.3. Costs and benefits

The goal of a financial analysis is to find out if the benefits outweigh the costs. The costs of a windpump system are:
- production and installation
- storage tank
- maintenance and reparations

The costs depend on:
- number of windpumps (the cost of a windpump is lower when the number is higher)
- materials
- transport
- depreciation of equipment
- number and price of manhours
- capacity of the storage tank

The optimum capacity of the storage tank can be calculated when we know:

- the time distribution of the windspeed (lit. 1 and 5)
- the demanded water quantity
- the chosen design windspeed of the windpump

When we choose a high design windspeed the water output will be high too, but also the starting windspeed. That makes the periods without output longer. A storage tank is necessary to bridge the periods without output. With a high design windspeed the storage tank needs a big capacity and that makes it expensive. Studies are made to find out how to choose the optimum design windspeed, see lit. 10 and 14. The conclusion is that the optimum design windspeed is more or less equal to the average windspeed of the site.

The benefits are difficult to calculate. It is possible to predict the water output when there are wind data available. But more difficult is to find out for example what the agricultural benefits are when the windpump is used for irrigation. It depends on the type of crops, the availability of water in critical months, the seasonal variation in the available water quantities, the evaporation, etc.
Benefits depend also on the life of the windpump which depend on the quality of the design, the production and the maintenance. Because the design is still new it is difficult to predict the life.

CONCLUSIONS
=====

The Diever 450 can be manufactured in a simple workshop, like the 12PU500. The materials are used efficiently, easily available and cheap. The dimensional accuracy of the design is low for most parts. The windmill is transportable on a roof-rack or a car trailer.

The design tipspeed ratio is lower (one in stead of two of the 12PU500) because it results in less wear of the piston and the pumpcylinder and lower gyroscopic moment of the rotor.
The disadvantage of the rotor is a lower $C_{p,max}$-value than the $C_{p,max}$ of the 12PU500, 0,27 in stead of 0,32. The value of 0,27 has to be checked by a windtunnel test. On the other hand the C_{Qstart} is higher, 0,28 in stead of 0,125 which results in a better starting behaviour.

The design windspeed is limited by the admissible pumprod forces (maximum 4,3 m/s for the 8" and 4,2 for the 5" p.v.c. pump). For the rotor design we chose: v_d= 3 m/s.

The rotor construction appears to be strong enough. Till now there are no cracked blade tips and blade supports on the prototype.

The inclined hinged vane system seems to be a better safety system than the hysteresis system. Off course it has to be tested on the prototype to find out its behaviour (δ-v curve) and the influence of the friction in the yaw bearings on the behaviour. The yaw bearings are made of impregnated wood which need less lubrification but has more friction than for instance ball-bearings.

The head cannot lift from the towerpipe because of a ring fixed on the towerpipe above the upper yaw bearing.

The four bars mechanism of the transmission replaces the crank/connecting-rod/cross-head mechanism, so the problems

with the cross-head are solved. the crank of appendix 6 is partly removable. The transmission with wooden bearings can be cheaper than with ball-bearings, but it is necessary that all the bearings are axially parallel. This makes the manufacturing rather difficult. With self-aligning ball-bearings as in appendix 7 the accuracy of axial parallelism can be less.

The acceleration of the piston in its lowest position is lower with the four bars mechanism.

The peakforces in the pumprod can be lower by:
- a lower design windspeed
- a lower $(A_p-A_{pr})/(A_{rm}-A_{pr})$ ratio. This can be achieved by:
  * a longer stroke $\rightarrow (A_p-A_{pr})$ smaller
  * larger rising mains $\rightarrow (A_{rm}-A_{pr})$ larger
- a lower spring stiffness of the pump system. This can be achieved by airchambers.

The pump of the Diever 450 has no airchambers. Therefore the construction is simple. The disadvantage is that the design windspeed is limited because the peakforces may not overstep the admissible forces, see fig. 5.2.

The number of strokes for the Diever 450 is more or less 6,5 million pro year at an average windspeed of 3 m/s. The wear of the 6" p.v.c.pump was small after 3,37 million strokes on the test rig. But a longer life test is advisable to test the wear after two years pumping at $\bar{v}$= 3 m/s (13 million strokes).

The tower is 10 m high. The tower construction is strong enough according to the requirements and calculations in chapter 6.

The windmill can be installed by hand. The use of a linch makes the installation easier, safer and faster than the method of the 12PU500 (see appendix 8).

The rated power output is more or less 250 W at v= 11 m/s and a rotation speed of 60 rpm, neglecting the friction in the yaw bearings. In chapter 5 we chose n= 60 rpm at v= 9 m/s and the rotor perpendicular in the wind for the fatigue strength

calculations. According to fig. 7.2 the rotation speed is 57 rpm at v= 9 m/s. The fatigue strength calculations are made 'on the safe side'.

RECOMMENDATIONS

The windpump Diever 450 isn't been tested completely. More research is advisable before large number of these windpumps can be produced. Recommendations for more research are:
- Windtunneltests. In a windtunnel tests can be done with a small model of the Diever 450 to measure:
    * the tipspeed ratio with and without load
    * the behaviour of the safety system by measuring the angle δ at different windspeeds
- Improving the starting behaviour by a contra weight and a leak hole in the piston.
- Design of a transmission which gives a longer stroke length than 250 mm. This can be done in combination with a design of a rotor eccentric from the towerpipe in stead of the side vane. It means a completely different head construction. The behaviour of a safety system with a eccentric rotor is more smooth than with a side vane. A longer stroke needs a smaller piston area to keep the same swept volume of the pump. Therefore the pumprod forces will be smaller, see formula 5.2.
- Changing the prototype according to the design changes as aforesaid and measuring the relation between the rotation speed-windspeed and the output characteristic
- Improving the safety system on the prototype to approach the ideal behaviour, see the graph on the next page.
- Lowering the spring stiffness of the pump system by a more elastic piston valve.
- The WOT is trying to find more possibilities for the material choice of the pump cylinder. Tests are done with p.v.c cylinders which resulted in a wear of 0,2 till 0,5 mm after 3,37 million strokes (par. 5.6). The experiences with cylinders of brass are better than with p.v.c. but these are more difficult to obtain. Sheets of brass are easier to obtain. Therefore the WOT is testing a pump with a pump cylinder made of iron with inside a 2 mm brass sheet glued against the wall with araldite. The WOT expects to have results of these tests at the end of this year.
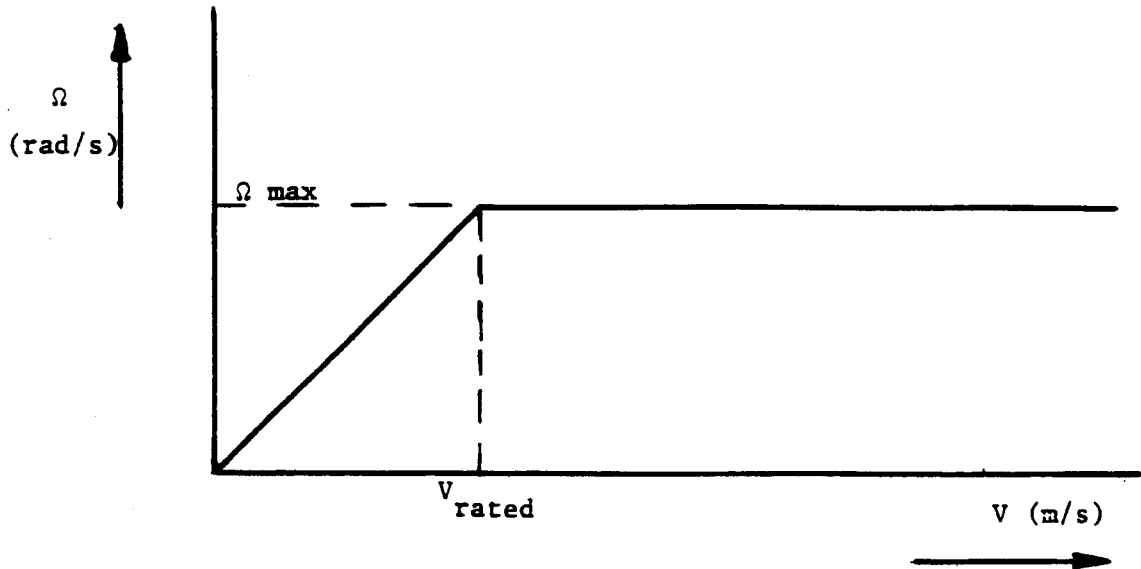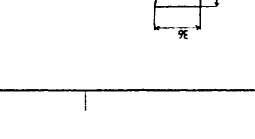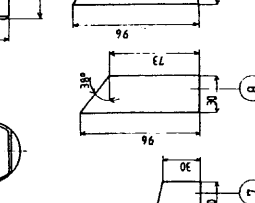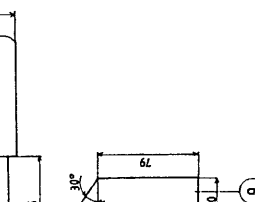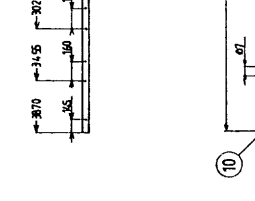
ideal v-Ω characteristic

- The Diever 450 can be too expensive for farmers with less than 1-1,25 ha land. The WOT is working on a smaller windpump, the 4PU250, which is much cheaper and simpler to produce.
- Before starting to introduce windmills it is very important to do first a feasibility study. This study must contain a costs-benefits analysis and a comparison to other pumpsystems like dieselpump, kerosinepump, solarpump, etc.

LITERATURE

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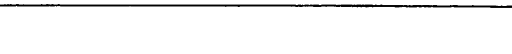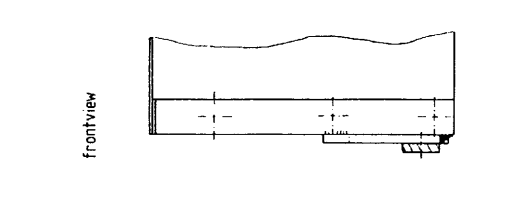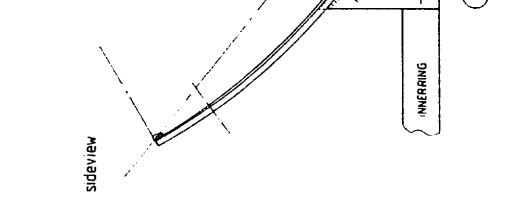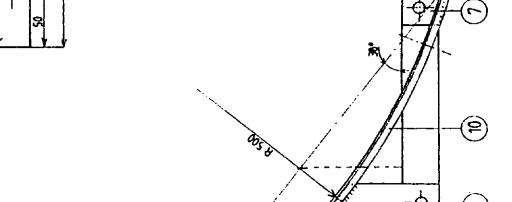16. 'Rotor design, part two' by A. Kragten, CWD, 1989

Appendix 1:

drawings Diever 450

DRAWING NR. 2

DRAWING NR. 3

DRAWING NR. 4

DRAWING NR.6C

DRAWING NR. 3B

DRAWING NR.6D

DRAWING NR. 6

DRAWING NR.6B

DRAWING NR. 6E

DRAWING NR. 5

PISTON PUMP

frontview

SCALE 1:60

cross section A-A (sideview)

backview

sideview

frontview

INNERRING CONNECTION

sideview

(OUTERRING CONNECTION IS EQUAL, ONLY WITHOUT PART Nr. ⑥)

Nr.⑥ AT THE INNER-SIDE OF THE RING

REMARK ALL WELDS ARE FILLET WELDS ∠3MM

SCALE 1:20
ø9 HOLES

frontview

sideview

sideview

frontview

frontview

OUTERRING

sideview

INNERRING

THE BLADE SUPPORTS ARE FIXED AT THE OUTERSIDE OF THE RING

TO FOLD 10MM

Part Number		Part name	Material or semi manufacture	Dimension in mm	Remarks
21	1 6	SPOKE CONNECTION	Flat iron Fe 37	30x6	L.85
20	76	BOLT, RING, NUT		M6 x 25	Class 4.6
19	108	BOLT, RING, NUT		M6 x 16	Class 4.6
18	8	BOLT, RING, NUT		M16 x 140	Class 4.6
17	7	BOLT, RING, NUT		M16 x 80	Class 6.6
16	1	ROTORSHAFT	Shaft Fe 52	ø50	L.1000
15	2	CLAMP	Angle iron Fe37	50 50 5	L.120
14	2	CLAMP	Angle iron Fe37	50 50 5	L.120
13	2	ANGLE IRON	Angle iron Fe37	40 40 4	L.300
12	6	SPOKE	Angle iron Fe37	40 40 4	L.1900
11	4	FLAT IRON	Flat iron Fe37	30x6	L.120
10	36	BLADE SUPPORT	Flat iron Fe37	30x6	L.390
9	18	BLADE SUPPORT	Flat iron Fe37	30x6	L.100
8	18	BLADE SUPPORT	Flatiron Fe37	30x6	L.96
7	36	BLADE CONNECTION	Flatiron Fe37	30x6	L.36
6	6	RING CONNECTION	Flatiron Fe37	30x6	L.1180
5	6	SPOKE	Flat iron Fe 37	30x6	L.1950
3	2	INNER RING	Flatiron Fe 37	30x6	L.3870
2	1	OUTER RING	Flatiron Fe 37	30x6	L.5940
1	18	BLADE	1mm sheet	400 x1000	

A4

ROTOR DIEVER 450

TECHNISCHE HOGESCHOOL TWENTE

DRAWING NR. 2

G.Wibenga 12-03-90

1:6

SCALE 1:22

THE TWO ROTORHALVES

SHEET IS ROLLED IN A BENDING -
RADIUS OF APPR. 500mm.

PART No ①

IMPORTANT

ROTORFRAME IS COMPOSED ON
ITS SHAFT TO PREVENT
"JUMPING AND DANCING" OF
THE SPOKES AND RINGS.
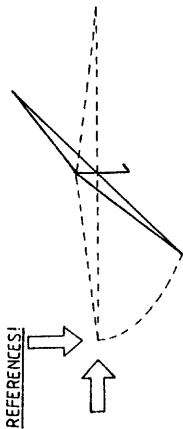
REFERENCES!
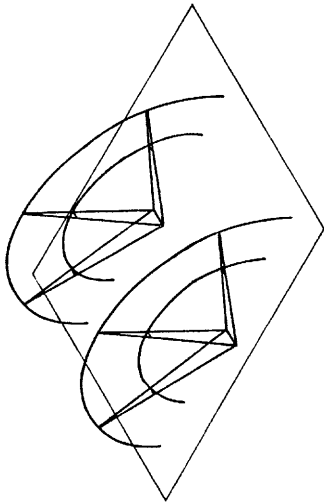
ORDER FOR WELDING OF THE
SPOKES SEE NUMBERING 1-6
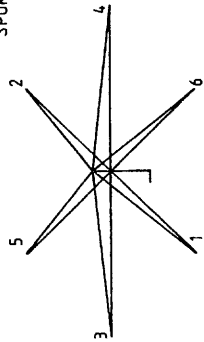
RINGS ARE POSITIONED,
CENTERED AND WELDED.

PARTS No 21 ARE WELDED.

POS.		BENAMING	AAN-TAL	MATERIAAL	TEK.NR. CODENR.	OPMERKING
	wijziging:			niet geol. maas ± 0.5 m.m.	proj.:	get.G.Wijbenga
		DRAWING NR. 2 A			schaal: 1:88	ges.20-03-'90
		ROTOR DIE VER 450				A4

TECHNISCHE HOGESCHOOL TWENTE

cross section E-E

view C

cross section D-D

Part nr		Part name	Material or semi manufacture	Dimension in mm	Remarks
44	2	FLAT IRON	Flat iron Fe 37	30×6	L.40
43	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.334
42	2	BOLT, RING, NUT		M8-50	Class 4.6
41	2	BOLT, RING, NUT		M6×16	Class 4.6
40	1	BUMPER	Rubber	60×60	thickness10 mm
39	4	YAW BEARING HOUSE	Angle iron Fe37	40×40×4	L.116
38	4	BOLT RING NUT		M12×100	Class 4.6
37	2	AXIAL BEARING CLAMP	Angle iron Fe 37	40×40×4	L.45
36	8	RIB	Sheet Fe37	27×27	L.90
35	4	AXIAL BEARING CLAMP	Angle iron Fe 37	30×30×3	thickness 4 mm
34	2	AXIAL BEARING PLATE	Sheet Fe37	Ø100	L.90
33	1	YAW BEARING HOUSE	Angle iron Fe 37	40×40×4	L.560
32	1	TOWERPPE RING	Flat iron Fe37	30×6	L.370
31	4	BOLT,RING,NUT		M10×20	Class 4.6
30	1	ANGLE IRON	Angle iron Fe37	40×40×4	L.466
29	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.900
28	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.374
27	2	ANGLE IRON	Angle iron Fe 37	40×40×4	L.940
26	2	SHAFT BEARING SUPPORT	Angle iron Fe 37	60×60×6	L.240
25	3	SIDE VANE CONNECTION	Flat iron Fe 37		L.78
24	4	BOLT,RING,NUT		M16×150	
23	2	YAW BEARING HOUSE	Angle iron Fe37	40×40×4	L.196
22	1	FLATHEAD PIN		Ø10	L.60
21	2	BOLT,RING,NUT		M12×120	Class 4.6
20	1	AXIAL BEARING RING	Sheet Fe 37	Ø60×8	thickness 6 mm
19	2	HINGEBEARING SUPPORT	Angle iron Fe 37	40×40×4	L.653
18	1	HINGESHAFT	Gaspipe Fe 37	1 1/2"	L.718
17	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.435
16	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.723
15	2	HINGE BEARING	Hardwood	100×60	L.204
14	2	FLAT IRON	Flat iron Fe37	30×6	L.204
13	3	YAW BEARING	Hardwood	198×188	thickness 56mm
12	1	AXIAL BEARING RING	Sheet Fe37	Ø60×8	thickness 6 mm
11	2	HINGECONNECTION	Flat iron Fe 37	30×6	L.40
10	2	BUMPER PLATE	Sheet Fe 37	60×60	thickness 4 mm
9	1	BUMPER SUPPORT	Rubber		L.40
8	1	BUMPER	Angle iron Fe 37	40×40×4	L.640
7	2	BUMPER SUPPORT	Angle iron Fe37	40×40×4	L.138
6	1	ANGLE IRON	Angle ron Fe37	40×40×4	L.382
5	2	ANGLE IRON	Angle iron Fe37	60×60×4	L.487
4	2	SHAFT BEARING SUPPORT	Angle iron Fe37	60×60×6	L.396
3	1	ROTORSHAFTBEARING	Hardwood	120×70	L.240
2	1	FLAT IRON	Flat iron Fe 37	30×6	L.240
1		Part name	Material or semi manufacture	Dimension in mm	Remarks

DRAWING NR. 3

HEAD DIEVER 450

UNIVERSITEIT TWENTE

G.Wijbenga

3-5-1990

scale 1:?

A4

cross section A-A

crosssection B-B

cross section F-F

REMARK:ALL WELDS ARE FILLET WELDS a3MM

DRAWING NR 3A

HEAD DIEVER 450

G.Wijbenga

26-4-1990

scale 1:6

A4

UNIVERSITEIT TWENTE

cross section A-A SCALE 1:3

detail C SCALE 1:6

detail B SCALE 1:6

detail D SCALE 1:6

detail E SCALE 1:6

detail F SCALE 1:6

vane-hinge connection SCALE 1:6

cross section K-K SCALE 1:3

detail G SCALE 1:3

detail H SCALE 1:3

detail J SCALE 1:3

REMARK ALL THE HOLES FOR NR 1 ARE ⌀9MM
ALL WELDS ARE FILLET WELDS ⌀3MM

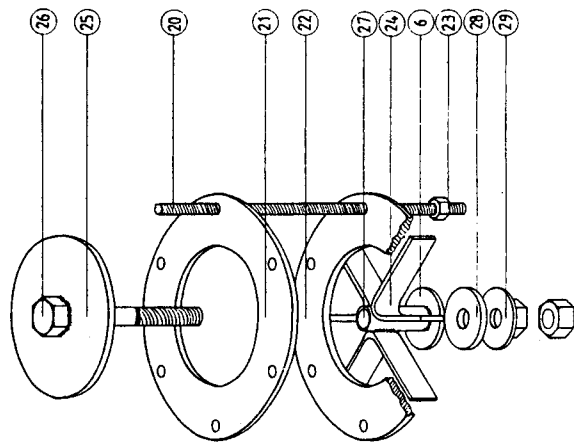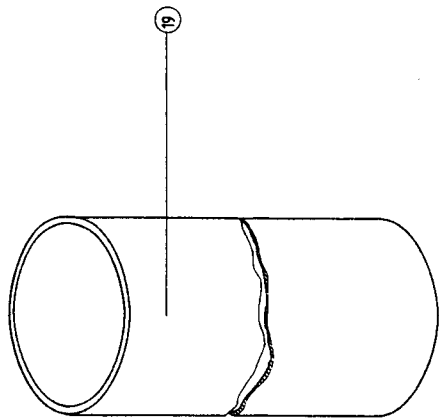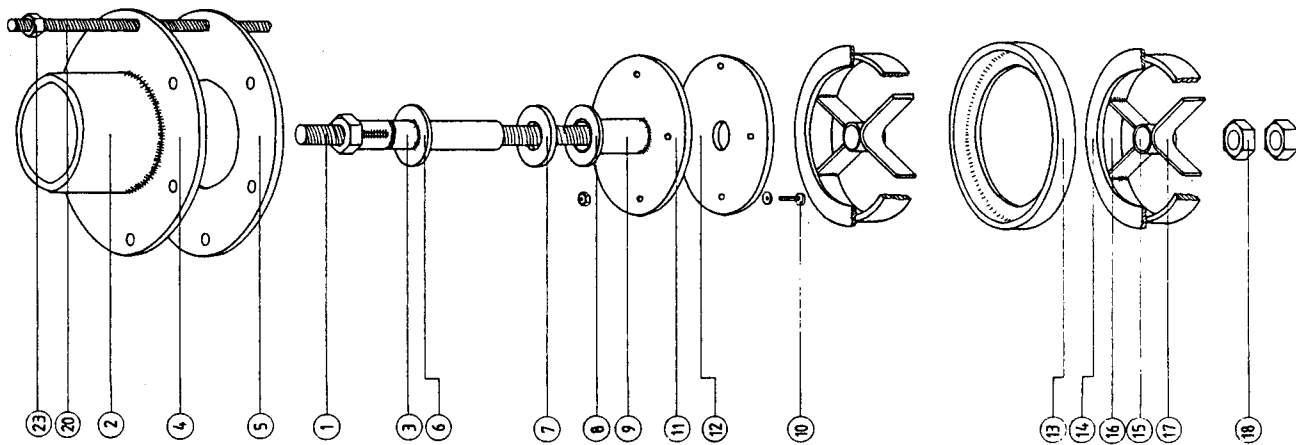| Part Num nr.		Part name	Material or Semi manufacture	Dimension in mm	Remarks
21	1	ARM SUPPORT	Gaspipe Fe 37	3/4"	L.1970
20	2	SUPPORT CONNECTION	Flat iron Fe 37	30×6	L.50
19	1	ARM SUPPORT	Gaspipe Fe37	1/2"	L.1970
18	1	VANE ARM	Gaspipe Fe37	3/4"	L.3600
17	5	ARM CONNECTION	Flat iron Fe 37	30×6	L.80
16	5	BOLT,RING,NUT	M8×20		Class 4.6
15	5	VANE CLAMP	Flat iron Fe37	30×6	L.175
14	2	VANE SUPPORT	Flat iron Fe37	30×6	L.600
13	1	SIDE VANE	Sheet Fe37	670×670	thickness 1 mm
12	2	BOLT, RING, NUT	M12×20		Class 4.6
11	1	ANGLE IRON	Angle iron Fe 37	30×30×3	L.590
10	1	ANGLE IRON	Angle iron Fe 37	30×30×3	L.780
9	1	ANGLE IRON	Angle iron Fe 37	40×40×4	L.1750
8	1	ANGLE IRON	Angle iron Fe 37	30×30×3	L.670
7	1	MAIN VANE	Sheet Fe 37	1000 ×1000	thickness 1 mm
6	1	ANGLE IRON	Flat iron Fe 37	30×6	L.900
5	1	ANGLE IRON	Angle iron Fe37	40×40×4	L.3910
4	1	VANE SUPPORT	Flat iron Fe37	30×6	L.900
3	1	MAIN VANE	Sheet Fe 37	2000 ×1000	thickness 1 mm
2	1	ANGLE IRON	Angle iron Fe.37	40×40×4	L.790
1	21	BOLT,RING,NUT	M 8×15		Class 4.6

DRAWING NR. 3B

MAIN VANE & SIDE VANE
DIE VER 450

G.Wijbenga 15-5-1990

scale 1:30

A4

TECHNISCHE HOGESCHOOL TWENTE

cross section A-A

cross section B-B

cross section C-C

side view

upper view

cross section D-D

view E

upper view

side view

side view rods mechanism

upper view assembly

side view assembly

REMARKS:

ALL BEARING BUSHES ARE PRESSED INSIDE THE HOUSING

THE PLAY BETWEEN BEARING BUSH AND PIPE MUST BE 0.1–0.2 MM.

ALL WELDS ARE FILLET WELDS △ 3 MM

Part num nr ber	Part name	Material or semi-manufacture	Dimension in mm	Remarks
22	1	CRANK PIN	Fe 37	Ø30	L.124
21	1	PUMPROD	Gaspipe Fe 37	1/2"	L.275
20	1	SHEET	Sheet	445×85	thickness 4 mm
19	1	BOLT	Gaspipe Fe 37	M10×20	Class 4.6
18	1	CONNECTING ROD	Gaspipe Fe 37	1"	L.275
17	1	THREADROD + 2 NUTS	Fe 37	M20	L.206
16	1	SEE DRAWING NR.			
15	1	RING	Sheet Fe 37	Ø60×11	thickness 4 mm
14	5	FLATIRON	Flat iron	30×6	L.85
13	3	TUMBLER SHEET	Sheet Fe 37	300×500	thickness 4 mm
12	4	GREASENIPPLE		M 6	
11	2	THREADROD + 2 NUTS	Fe 37	M20	L.140
10	1	CRANK SHEET	Sheet Fe 37	137×50	thickness 6 mm
9	1	CRANK SHEET	Sheet Fe 37	145×50	thickness 6 mm
8	1	CRANK FLAT IRON	Flatiron Fe 37	30×6	L.370
7	2	RING	Sheet Fe 37	Ø60×30	thickness 4 mm
6	2	BEARING BUSH	Hardwood	Ø60	L.35
5	1	BEARING PIPE	Gaspipe Fe 37	3/4"	L.164
4	1	BEARING HOUSING	Gaspipe Fe 37	2"	L.135
3	3	BEARING HOUSING	Gaspipe Fe 37	2"	L.60
2	6	BEARING BUSH	Hardwood	Ø60	L.35
1	1	BEARING PIPE	Gaspipe Fe 37	3/4"	L.85

sch. GWijbenga

datum 2-4-1990

schaal 1:5

DRAWING NR. 4

TRANSMISSION DIEVER 450

UNIVERSITEIT TWENTE

A4

Parts list

Part number	Number	Part name	Material or semimanufact.	Dimension	Remark
29	1	RING	Sheet Fe37	⌀50×16	3 mm thick
28	1	SHOCKRING	Leather	⌀50×18	5 mm thick
27	1	GUIDING	Gaspipe Fe37	1/2"	50 mm long
26	1	VALVE ROD	Bolt	M16×110	4.6 class
25	1	FOOT VALVE	Sheet Fe37	⌀130×16	6 mm thick
24	3	RIB	Flatiron Fe37	20×5	160 mm long
23	12	NUT		M10	class 5
22	1	BOTTOM FLANGE	Sheet Fe37	⌀200×114	6 mm thick
21	1	BOTTOM GASKET	Leather	⌀200×114	5 mm thick
20	6	THREAD ROD	Fe 37	M10	600 mm long
19	1	PUMPCYLINDER	PVC pipe	⌀160×150,6	560 mm long
18	5	NUT		M16	klasse 5
17	4	RIB	Flatiron Fe37	20×5	110 mm long
16	2	RING	Flatiron Fe37	20×5	400 mm long
15	2	PIPE	Gaspipe Fe37	1/2"	26 mm long
14	2	SEAT	Sheet Fe37	⌀146×114	4 mm thick
13	1	PISTON CUP	Leather	⌀150×114	5 mm thick 15mm high
12	1	PISTON VALVE	Leather	⌀130×24	5 mm thick
11	1	PISTON VALVE	Sheet Fe37	⌀130×27	6 mm thick
10	4	BOLT.NUT.RING		M4×15	4.6 class
9	1	GUIDING	Gaspipe Fe37	3/4"	50 mm long
8	1	RING	Sheet Fe37	⌀50×27	3 mm thick
7	1	SHOCKRING	Leather	⌀50×24	5 mm thick
6	2	RING	Sheet Fe37	⌀50×22	3 mm thick
5	1	UPPER GASKET	Leather	⌀200×90	5 mm thick
4	1	UPPER FLANGE	Sheet Fe37	⌀200×90	6 mm thick
3	1	PISTON ROD	Gaspipe Fe37	1/2"	300 mm long
2	1	SOCKET	Fe 37	3"	
1	1	THREAD ROD	Fe 37	M16	420 mm long

Balloon references on the drawing: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21, 22, 23, 24, 25, 26, 27, 28, 29.

This page is a full-page engineering drawing.

Part nr	Part name	Material or semimanufact.	Dimension	Remark	Number
29	RING	Sheet Fe37	Ø50×16	3 mm thick	1
28	SHOCKRING	Leather	Ø50×18	5 mm thick	1
27	GUIDING	Gaspipe Fe37	1/2"	50 mm long	1
26	VALVE ROD	Bolt	M16×110	4.6 class	1
25	FOOT. VALVE	Sheet Fe37	Ø85×16	6 mm thick	1
24	RIB	Flatiron Fe37	20×4	120 mm long	2
23	NUT		M10	klass 5	12
22	BOTTOM FLANGE	Sheet Fe37	Ø160×72	6 mm thick	1
21	BOTTOM GASKET	Leather	Ø160×72	5 mm thick	1
20	THREAD ROD	Fe 37	M10	600 mm long	6
19	PUMPCYLINDER	PVC pipe	Ø110×101.6	560 mm long	1
18	NUT		M16	class 5	5
17	RIB	Flatiron Fe 37	20×4	60 mm long	4
16	PIPE	Gaspipe Fe37	1/2"	26 mm long	2
15	RING	Ratiron Fe 37	20×4	265 mm long	2
14	PISTONCUP	Leather	Ø101×72	5 mm thick	1
13	SEAT	Sheet Fe 37	Ø 97×72	4 mm thick	2
12	PISTON VALVE	Leather	Ø85×24	5 mm thick	1
11	PISTON VALVE	Sheet Fe 37	Ø85×27	6 mm thick	1
10	BOLT,NUT,RING		M4×15	4.6 class	4
9	GUIDING	Gaspipe Fe37	3/4"	50 mm long	1
8	RING	Sheet Fe37	Ø50×27	3 mm thick	1
7	SHOCKRING	Leather	Ø50×24	5 mm thick	1
6	RING	Sheet Fe 37	Ø50×22	3 mm thick	2
5	UPPER GASKET	Leather	Ø160×64	5 mm thick	2
4	UPPER FLANGE	Sheet Fe37	Ø160×64	6 mm thick	1
3	PISTON ROD	Gaspipe Fe37	1/2"	300 mm long	1
2	SOCKET	Fe 37	2"		1
1	THREAD ROD	Fe 37	M16	420 mm long	1

Universiteit Twente
faculteit der werktuigbouwkunde

ONTWERP EN CONSTRUCTIELEER

4" PISTON PUMP cross-section assembly

Schaal 1:2,2
Getekend F. BRUGHUIS
Datum 13-11-89

DRAWING NR.5C

A4

REMARKS:

V	V-weld
K	K-weld
◿	fillet weld
◿	fillet weld at both sides
⌀	to weld all round

DRAWING NR. 50

4" PISTON PUMP
parts

UNIVERSITEIT TWENTE

Cross section B-B

Cross section A-A

A4

detail F   scale 1:3

Part number		Part name	Material or semi manufacture		Dimension in mm	Remarks
37	8	BOLT, 2 NUTS			M16×65	Class 4.6
36	8	PIPE	Gaspipe	Fe 37	1/2"	L.30
35	4	TOWERPIPE SUPPORT	Flat iron	Fe 37	30×6	L.140
34	8	SHEET	Sheet	Fe 37	40×10	Thickness 6 mm
33	1	YAW BEARING SEAT	Sheet	Fe 37	φ190×115	L.535
32	1	RING	Flat iron	Fe 37	30×6	Thickness 6 mm
31	4	TOWERPIPE	Gaspipe	Fe 37	4"	L.1625
30	4	HORIZONTAL	Angle iron	Fe 37	30×30×3	L.426
29	2	STEP	Angle iron	Fe 37	30×30×3	L.325
28	2	STEP	Angle iron	Fe 37	30×30×3	L.210
27	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.953
26	8	HORIZONTAL	Angle iron	Fe 37	30×30×3	L.692
25	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.980
24	2	STEP	Angle iron	Fe 37	30×30×3	L.480
23	2	STEP	Angle iron	Fe 37	30×30×3	L.805
22	2	STEP	Angle iron	Fe 37	30×30×3	L.910
21	2	STEP	Angle iron	Fe 37	30×30×3	L.670
20	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.380
19	8	HORIZONTAL	Angle iron	Fe 37	30×30×3	L.1065
18	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.1269
17	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.1140
16	24	BOLT, RING NUT			M 8×20	Class 4.6
15	36	BOLT, RING NUT			M10×25	
14	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.1190
13	8	HORIZONTAL	Angle iron	Fe 37	30×30×3	L.923
12	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.1270
11	12	CONNECTING IRON	Angle iron	Fe 37	50×50×5	L.160
10	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.1370
9	8	CONNECTING IRON	Angle iron	Fe 37	30×30×3	L.160
8	8	HORIZONTAL	Angle iron	Fe 37	30×30×3	L.1212
7	8	DIAGONAL	Angle iron	Fe 37	30×30×3	L.2300
6	8	HORIZONTAL	Angle iron	Fe37	30×30×3	L.1000
5	4	LEG	Angle iron	Fe37	50×50×5	L.500
4	4	LEG	Angle iron	Fe 37	50×50×5	L.2000
2	4	LEG	Angle iron	Fe 37	50×50×5	L.2000
1	4	LEG	Angle iron	Fe 37	50×50×5	L.3000

DRAWING NR. 6

TOWER DIEVER 450   assembly

UNIVERSITEIT TWENTE

A4

scale 1:30   G.Wijpenga   9-4-1990

sideview

sideview

backview

upperview

backview

upperview

Ø11

detail E

detail D

detail C

view P

Cross section detail A

detail B

REMARK:

⊿ FILLET WELD

○ WELD ALL ROUND

REMARK: ALL WELDS ARE WELDED ALL ROUND △3

Part Num ber		Part name	Material or semi manufacture	Dimension in mm	Remarks
6	6	STAIRSSUPPORT	Angle iron Fe37	30 30 3	L.150
5	2	STAIRSSUPPORT	Angle iron Fe37	30×30×3	L.140
4	1	STEP	Angle iron Fe37	30×30×3	L.370
3	21	STEP	Angle iron Fe37	30×30×3	L.350
2	2	STAIRS LEG	Angle iron Fe37	30×30×3	L.6000
1	1	HORIZONTAL	Angle iron Fe37	50×50×5	L.3000

DRAWING NR. 6B

prod. scale 1:11
res. GWijbenga dat. 12-4-1990

A4

STAIRS DIEVER 450

TECHNISCHE HOGESCHOOL TWENTE

REMARK: FILLET WELDS ⌀ 3MM

POS.	NAME	NUMBER	MATERIAL	DIMENSIONS	REMARKS
18	FLAT IRON	1	Fe 37	30×6	L.565
17	SEAT	1	Wood	400×180	10mm thick
16	ANGLE IRON	1	Fe 37	30×30×3	L.440
15	ANGLE IRON	1	Fe 37	30×30×3	L.174
14	ANGLE IRON	1	Fe 37	30×30×3	L.440
13	FLAT IRON	1	Fe 37	30×6	L.600
12	ANGLE IRON	1	Fe 37	30×30×3	L.30
11	PLATFORM	1	Wood	370×246	10mm thick
10	BOLT / NUT	8		M8×25	4.6 class
9	ANGLE IRON	1	Fe 37	30×30×3	L.440
8	ANGLE IRON	1	Fe 37	30×30×3	L.240
7	ANGLE IRON	1	Fe 37	30×30×3	L.437
6	BOLT / NUT	1		M10×20	4.6 class
5	ANGLE IRON	1	Fe 37	30×30×3	L.400
4	ANGLE IRON	1	Fe 37	30×30×3	L.330
3	ANGLE IRON	1	Fe 37	30×30×3	L.750
2	ANGLE IRON	1	Fe 37	30×30×3	L.800
1	ANGLE IRON	1	Fe 37	30×30×3	L.260

DRAWING NR. 6C

SEAT DIEVER 450

UNIVERSITEIT TWENTE

DIMENSIONS 1:15   F Brughuis   6-7-90   A4

TO TWIST 90°

view Q (see drawing nr. 6)

scale 1:10
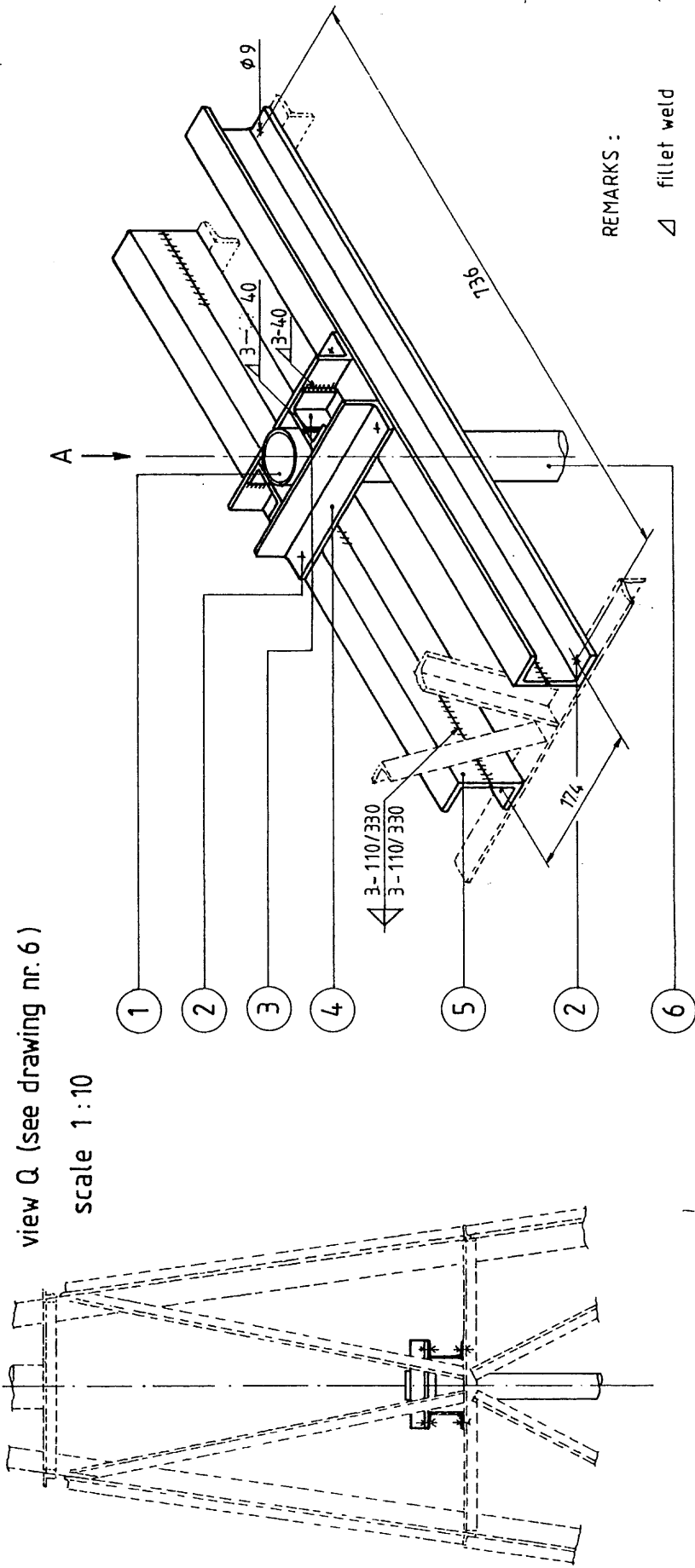
view A

REMARKS :

△ fillet weld

▽ fillet weld both sides
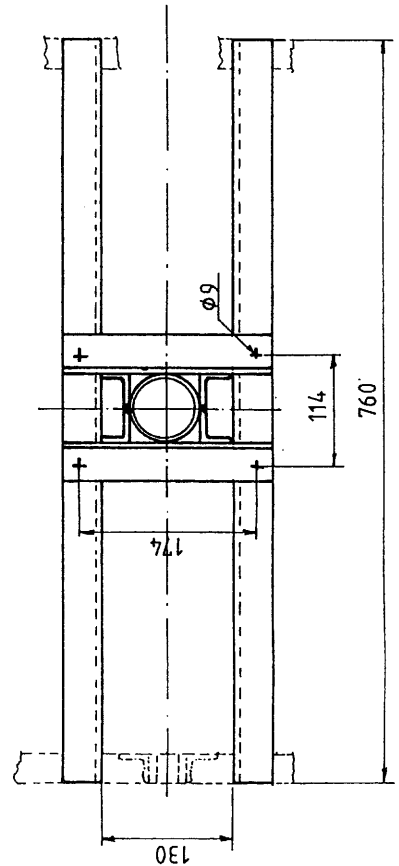
Stuk nr.	Aantal	Benaming	Materiaal en/of half fabrikaat	Normaanduiding of afmetingen	Opmerking
6	1	RISING MAIN	Gaspipe Fe37	2"	L.6000
5	4	SUPPORT	Angle iron Fe37	40×40×4	L. 760
4	2	ANGLE IRON	Angle iron Fe37	40×40×4	L. 210
3	4	ANGLE IRON	Angle iron Fe37	30×30×3	L. 40
2	8	BOLT / NUT		M8×20	class 4.6
1	1	SOCKET	Fe 37	2"	

niet getol. maten : 0.5 mm.

Titel: PUMP SUPPORT DIEVER 450

TECHNISCHE HOGESCHOOL TWENTE

AFD. W.B.
ONTWERP- EN CONSTRUCTIELEER

Schaal: 1:7,5   Datum: 2-7-90

Getekend: F. Brughuis

Gezien:

Behoort bij:

Tekening nr. DRAWING NR. 6D

A4

O & C /

scale 1:11

170

400

006

③

170

400

②

①

30

50 30

700

900

③

2930

①
②

900

006

④

3850

3010

3880

3000

10000

detail A

60°
40
R20

detail B

70
40
20
90
30

scale 1:75

A
30°
B
2000
2000
2000
3200
5800
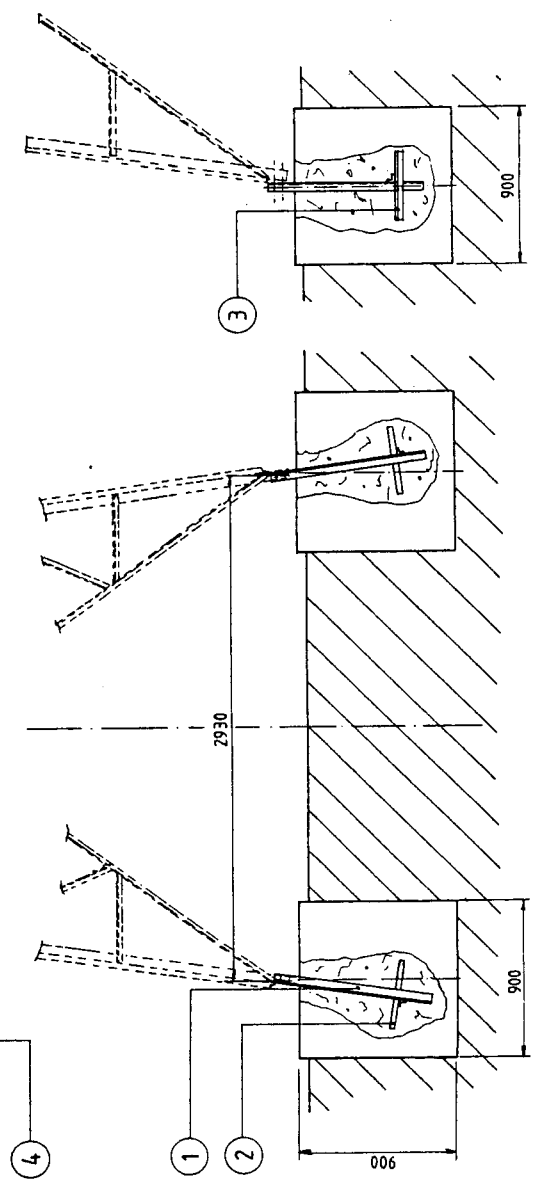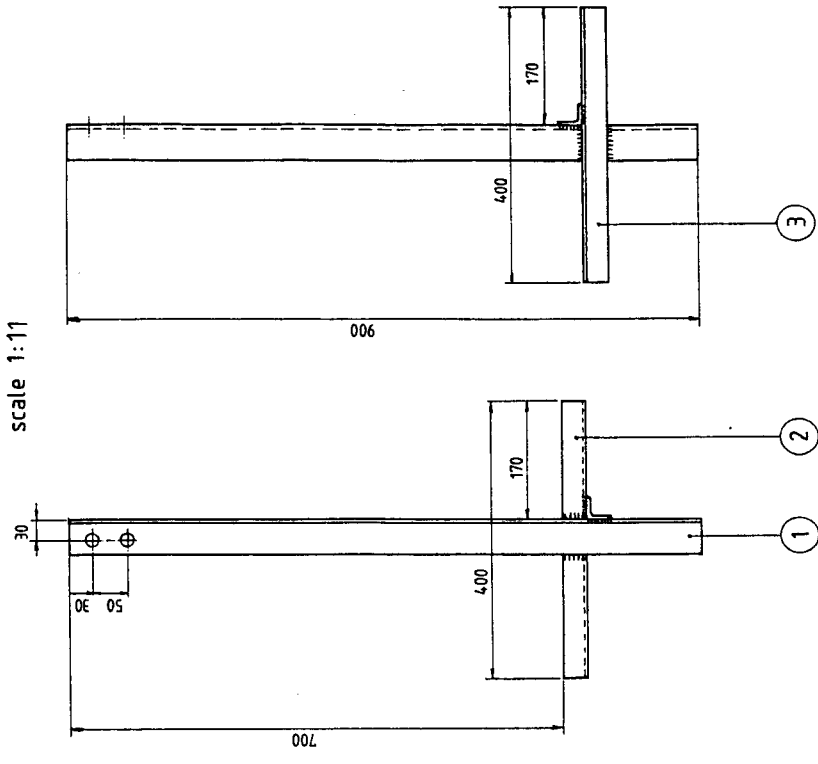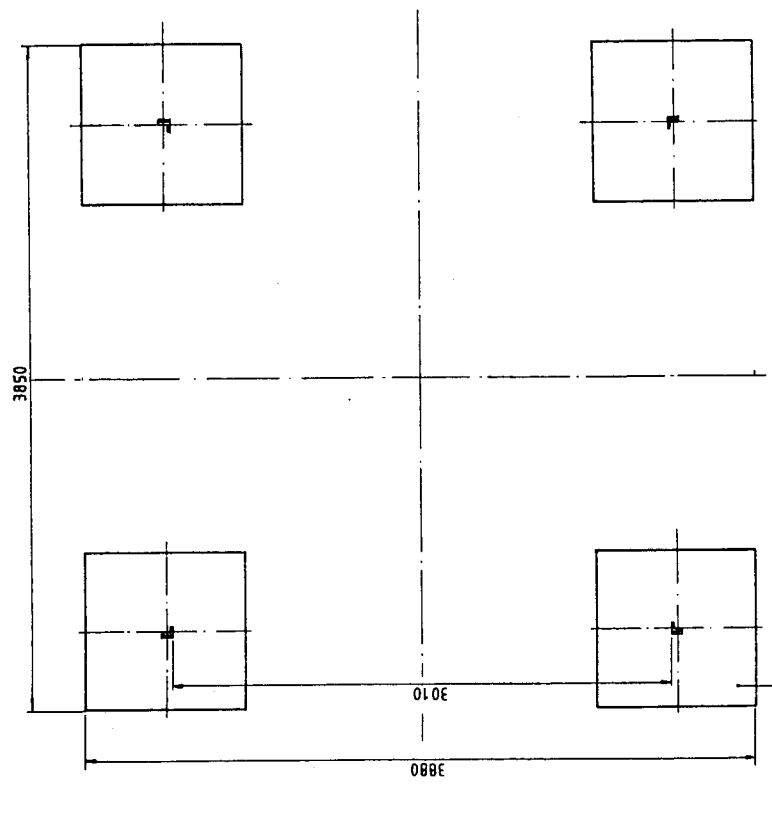
Part Num nr. ber		Part-name	Material--	Dimension	Remarks
7	2	FLAT IRON	Fe 37	60 × 6	L.70
6	2	BOLT, NUT	-	M20×30	class 4.6
5	2	FLAT IRON	Fe 37	60 × 6	L. 90
4	1	GAS PIPE	Fe 37	2"	L. 1920
3	2	GAS PIPE	Fe 37	3"	L. 6000
2	2	GAS PIPE	Fe 37	2 ½"	L. 500
1	1	GAS PIPE	Fe 37	½"	L. 300

DRAWING NR. 7

niet getol.
maten
±0.5 M.M.

proj. ⊕⊏⊐

Scale 1:3

get.:F. Brughuis

gez.:

dat.: 31-8-90

A4

CABLE SUPPORT YOKE

UNIVERSITEIT TWENTE

APPENDIX 2: The choice of the design tipspeed ratio

Matching the pump at a certain design windspeed we find in par. 5.4:

$$V_s = \frac{\rho_a * \pi^2 * C_p * \eta_{tot} * v^2 * D_r^3}{8 * \rho_w * g * H * \eta_{vol} * \lambda} \qquad \text{(formula 5.5)}$$

In par 5.2 we find that the maximum pumprod force (peakforce):

$$F_{peak} = k * \rho_w * H * (A_p - A_{pr}) * \{ g + \Omega^2 * R_c * (A_p - A_{pr}) / (A_{rm} - A_{pr}) \} \quad \text{(formula 5.2)}$$

The peakforce consists of different forces:

$F_{peak} = F_{stat} + F_a + F_s + F_w + F_{wl}$

F_{stat} = force caused by the static pressure

$\quad = \rho_w * g * H * (A_p - A_{pr})$

F_a = force caused by the acceleration of the water, piston and pumprod

$\quad = \{ (A_p - A_{pr}) * \rho_w * (\frac{A_p - A_{pr}}{A_{rm} - A_{pr}} * H + H_{cyl}) + m_p + m_{pr} \} * a_p$

F_s = force caused by the shock of the piston against the water column at the moment that the piston valve closes

$\quad = \{ \sqrt{(H_{cyl} * (A_p - A_{pr}) * \rho_w * c)} + \frac{A_p - A_{pr}}{A_{rm} - A_{pr}} * \sqrt{(H * (A_{rm} - A_{pr}) * \rho_w * c)} \} * v_p$

F_w = force caused by the friction of the piston and the cylinder

$\quad = (p_{stat} + p_a + p_s + p_{wl}) * \pi * D_p * h_{cup} * f = \frac{F_{peak} - F_w}{A_p - A_{pr}} * \pi * D_p * h_{cup} * f =$

$\quad = \frac{F_{peak} * \pi * D_p * h_{cup} * f}{A_p - A_{pr} + (\pi * D_p * h_{cup} * f)}$

F_{wl} = force caused by the conductor resistance in the pump and rising main (see lit. 9)

$\quad = (1 + \xi) * \frac{1}{2} * \rho_w * v_p^2 * (A_p - A_{pr})^3 / (A_{rm} - A_{pr})^2$

with:
H_{cyl} = length of the pump cylinder
m_p = mass of the piston
m_{pr} = mass of the pumprod
c = spring stiffness of the pump system
f = friction coefficient cup-pump cylinder
D_p = piston diameter
a_p = piston acceleration in its lowest position
v_p = piston velocity at the moment that the piston valve closes

ξ= conductor resistance of the pump installation

Two situations are considered:
- The 12PU500 with λ_d= 1
- The 12PU500 with λ_d= 2

Maximum pumprod forces are calculated for two windspeeds and the rotor perpendicular in the wind:
1. v= 9 m/s, which is considered as a fatigue load
2. v= 12 m/s, which is considered as the maximum, no fatigue load ('static load')

Next assumptions are made:
- The stroke length is 240 mm in both cases
- F_{wl} is negligible compared to the other forces
- If the spring stiffness remain equal for λ_d= 1 and λ_d= 2. Then the shockforce for λ_d= 1 is a little bit smaller than for λ_d= 2. For the calculations here we assume that F_s remain equal.

Calculations are made with the next data;
v_d = 3 m/s
H = 15 m
H_{cyl} = 0,4 m
A_{rm} = 2206 mm²
A_{pr} = 380 mm²
L_{pr} = length pumprod= 20 m
h_{cup} = 0,02 m
f = 0,15
λ(at v= 9 m/s) = 1,6 * λ_d
λ(at v= 12 m/s)= 1,7 * λ_d

The results of the calculations:
<u>1a. v= 9 m/s, λ_d= 1</u>
n= 60 rpm
$F_{stat,1}$= 1250 N
$F_{a,1}$= 2945 N
$F_{w,1}$= 0,103 * $F_{peak,1}$
$F_{s,1}$= 0,897 * $F_{peak,1}$ - 4195 N

1b. v= 12 m/s, $\lambda_d= 2$

n= 120 rpm

$F_{stat,2}= 625$ n

$F_{a,2}= 3369$ N

$F_{w,2}= 0,141 * F_{peak,2}$

$F_{s,2}= 0,859 * F_{peak,2} - 3994$ N

If $F_{s,1}= F_{s,2}$ then: $F_{peak,1}= 0,958 * F_{peak,2} + 224$ N

2a. v= 12 m/s, $\lambda_d= 1$

n= 80 rpm

$F_{stat,1}= 1250$ N

$F_{a,1}= 5236$ N

$F_{w,1}= 0,103 * F_{peak,1}$

$F_{s,1}= 0,897 * F_{peak,1} - 6786$ N

2b. v= 12 m/s, $\lambda_d= 2$

n= 160 rpm

$F_{stat,2}= 625$ N

$F_{a,2}= 5989$ N

$F_{w,2}= 0,141 \ F_{peak,2}$

$F_{s,2}= 0,859 * F_{peak,2} - 6614$ N

If $F_{s,1}= F_{s,2}$ then: $F_{peak,1}= 0,958 * F_{peak,2} - 143$ N

Conclusions:

The peakforces in the pumprod are almost equal for $\lambda_d= 1$ and $\lambda_d= 2$ at both windspeeds. Lowering the tipspeed ratio does not give a clear change of the peakforces. However a lower tipspeed ratio means less friction between the piston and the pump cylinder because the friction force F_w is smaller and the contact area (= $\pi * D_p * h_{cup}$) is larger. Less friction and less strokes result in less wear of the piston cup and the pump cylinder.

In our example:

- $F_{w,1} \quad 0,73 * F_{w,2}$ if $F_{peak,1}= F_{peak,2}$
- The contact area for $\lambda_d= 1$ is $\sqrt{2}$ times the contact area for $\lambda_d=2$
- when the wear is proportional to F_w and the number of strokes and inversely proportional to the contact area then the wear for $\lambda_d= 1$ is 26 % of the wear for $\lambda_d= 2$.

APPENDIX 3: Graphs of the position, the speed and acceleration of the piston as function of the time.
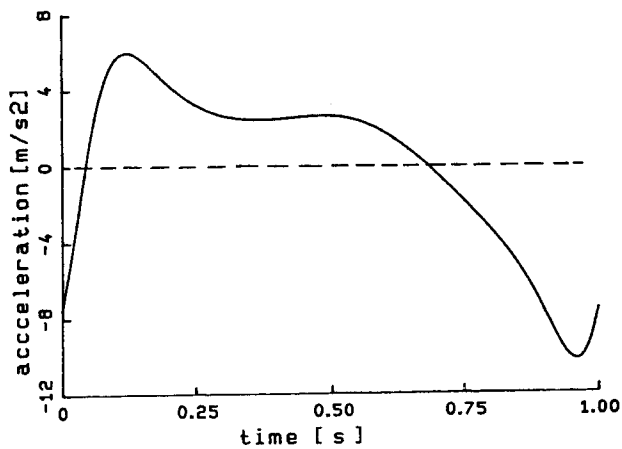
The rotation speed is 60 r.p.m.
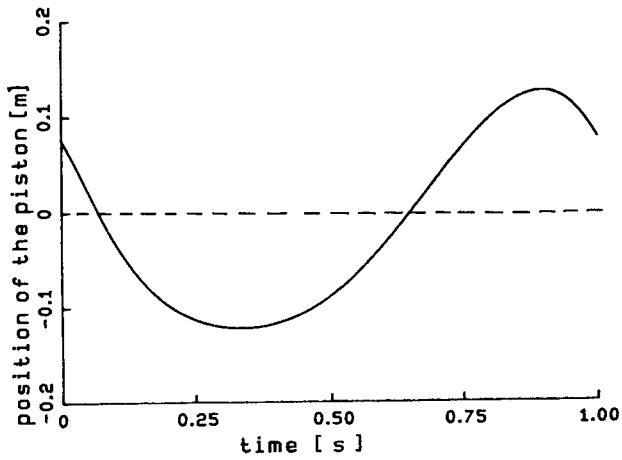
graph 1
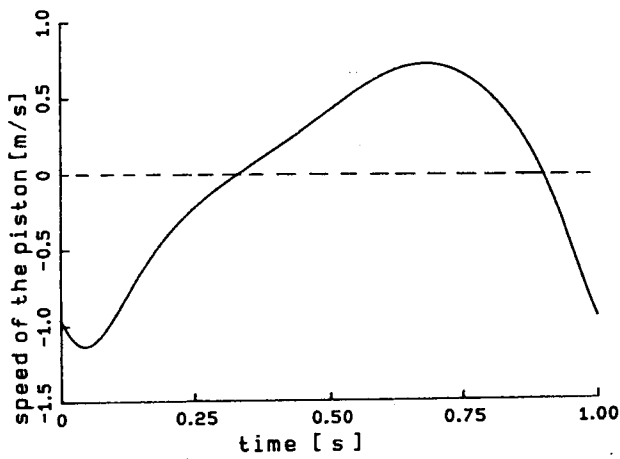
graph 2

graph 3

APPENDIX 5: Results of the measurements on the Diever 450 prototype

1. The 4" p.v.c. pump (ϕ110x105,2)

   stroke length s= 240 mm

   valve lifts are 8 mm

   waterlifting head H= 10,5 m

   rising mains of 2", pumprod of ½"

   F_{peak}= k * (855 + 0,522*n^2), with n in rpm.

n in rpm.	F_{peak} in N	k	n in rpm.	F_{peak} in N	k
21	1920	1,77	57	4340	1,70
27	2020	1,63	60	4340	1,59
30	2220	1,68	60	4440	1,62
31	2420	1,78	60	4540	1,66
40	2830	1,67	60	4650	1,70
41	3030	1,75	60	4850	1,77
44	3430	1,83	63	5450	1,86
46	3030	1,54	67	5050	1,58
50	3940	1,82	67	5450	1,70
54	3640	1,53	67	5760	1,80
54	3740	1,57	71	6260	1,80
54	4650	1,96	75	5250	1,38
55	3540	1,45	75	5760	1,52
57	3940	1,54	75	6570	1,73

The overshoot factor k is between 1,38 and 1,96. The average value is 1,68.

2. 6" p.v.c. pump (⌀160x150)

    stroke length s= 120 mm

    valve lifts are 8 mm

    waterlifting head H= 10,5 m

    rising mains of 2", pumprod of ½"

      $F_{peak} = k * (1790 + 1,14*n^2)$, with n in rpm.

n in rpm.	F_{peak} in N	k	n in rpm.	F_{peak} in N	k
26	3840	1,50	52	7270	1,49
30	4040	1,43	60	8280	1,40
39	4440	1,26	60	8480	1,44
40	4850	1,34	60	8690	1,47
40	6260	1,73	63	9090	1,44
41	5860	1,58	63	9090	1,44
46	5760	1,37	67	9898	1,43
46	6260	1,49	75	11620	1,73
48	6260	1,42	80	12120	1,33
48	6570	1,49	80	11920	1,31
50	6570	1,42			

The overshoot factor k is between 1,31 and 1,73. The average
value is 1,45.

APPENDIX 6: Crank with a variable length

As determined in par. 6.2.1 the maximum pumprod force (the peak force) F_{pr}= 9700 N
with: n= 80 rpm., k= 2,8, 4" pump, H=16 m, calculated as a static load.
For fatigue load we assume that: n= 60 rpm., k= 2,8. The pumprod force (the peak force) is then:
F_{pr}= 6900 N
The peak force causes the highest stresses in the crank. With the help of the photographs in appendix 3 we assume that the peak force appears when the piston is 8 mm above its lowest position, see figure 1.
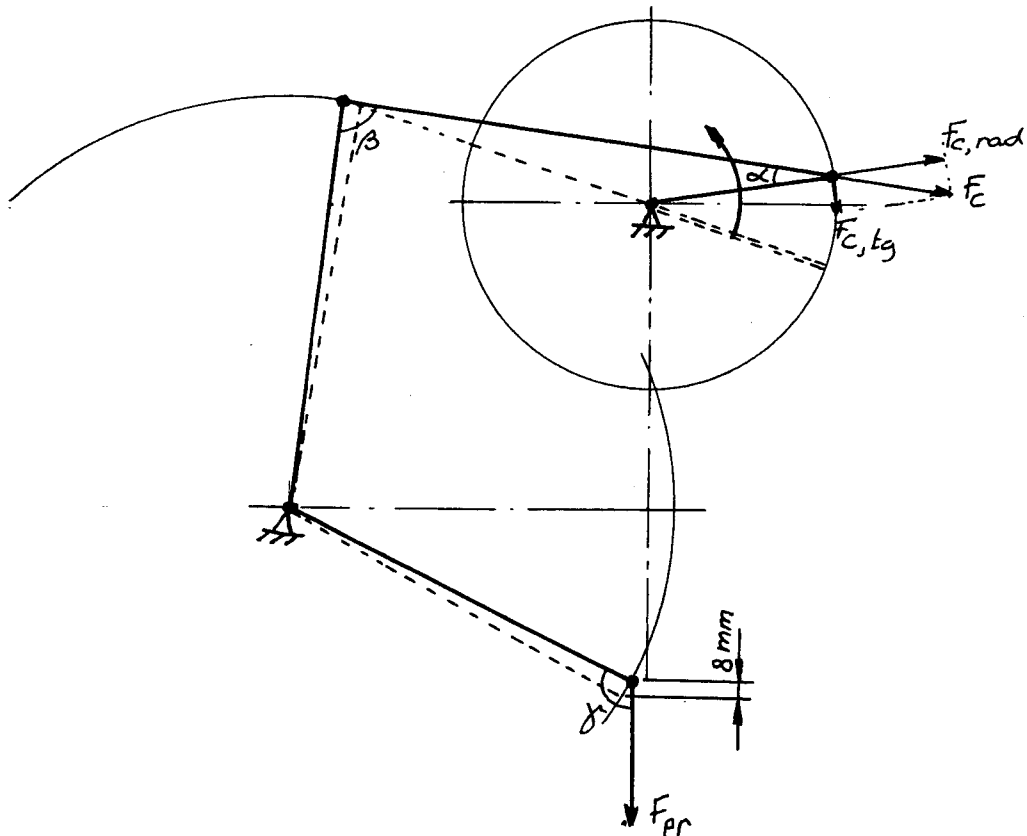
figure 1: situation at the moment of the peak force

The forces:
F_c= force on the crank shaft= $^{25}/_{27}*\sin\gamma *\sin\beta*F_{pr}$= $0,82*F_{pr}$
$F_{c,rad}$= radial force on the crank shaft= $F_c*\cos\alpha$
$F_{c,tg}$ = tangential force on the crank shaft= $F_c*\sin\alpha$

Data:

D_s = diameter rotor shaft   50 mm

D_{c1}= diameter crank shaft   30 mm

D_{c2}= diameter crank shaft (welde part)   50 mm

a = critical cross-section welds   5 mm

A_b = critical cross-section M10 bolt   52,3 mm²

α =   17°

β =   89°

γ =   117°

R_c = crank length   120 mm

b x t = dimensions crank plates= flat iron   80 x 12 mm

Calculations and results:

Forces on the crank shaft:

static load: F_c = 0,8 * 9700= 7950 N

            $F_{c,rad}$= 7950 * cos 17°= 7600 N

            $F_{c,tg}$ = 7950 * sin 17°= 2320 N

Fatigue load: F_c = 5660 N

            $F_{c,rad}$= 5410 N

            $F_{c,tg}$ = 1660 N

Stresses in the plates:
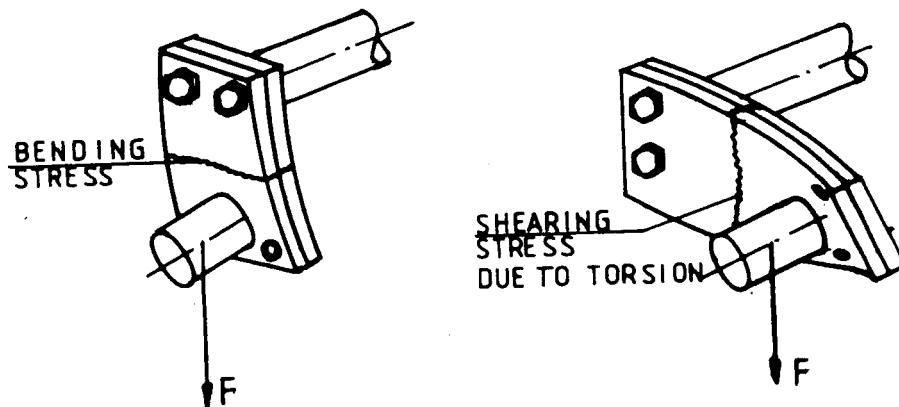
figure 2: load situations crank plates

Assumed is that each plate takes half the torsional and bending moments.

τ_t = shearing stress due to torsion= $\frac{1}{2} * M_t/W_t$= $\frac{1}{2} * \frac{F_{c,tg}*65}{\frac{1}{9}*b*t^2}$

σ_b = bending stress= M_b/W_b= $\frac{1}{2} * \frac{F_{c,rad}*651}{\frac{1}{6}*b*t^2}$

σ_e = equivalent stress= $\sqrt{(\sigma_b^2 + 3*\tau_t^2)}$

static load: σ_e= 138 N/mm² ($\bar{\sigma}$= 160 N/mm²)

fatigue load: σ_e= 99 N/mm² ($\bar{\sigma}$= 105 N/mm²)

Stresses in the welds:

At the crank side:

σ_b = bending stress= M_b/W_{bs}= $\dfrac{F_c * (65 - 12)}{0,1*\{(D_{c2}+2*a)^4 - D_{c2}^4\}/D_{c2}}$

static load: σ_b= 31 N/mm² ($\bar{\sigma}$= 160 N/mm²)

fatigue load: σ_b= 22 N/mm² ($\bar{\sigma}$= 35 N/mm²)

At the shaft side:

τ_t = shearing stress due to torsion= M_w/W_t= $\dfrac{F_{c,tg}*R_c}{\pi*D_s^2*a}$

σ_b = bending stress= M_b/W_b= $\dfrac{F_{c,rad} * (65 + 12)}{0,1*\{(D_s+2*a)^4 - D_s^4\}/D_s}$

σ_e = equivalent stress= $\sqrt{(\sigma_b^2 + 3*\tau_t^2)}$

static load: σ_e= 45 N/mm² ($\bar{\sigma}$= 160 N/mm²)

fatigue load: σ_e= 32 N/mm² ($\bar{\sigma}$= 35 N/mm²)

Stresses in the bolts:

τ_1 = shearing stress= $F_c/(4*A_b)$

τ_2 = shearing stress due to moment $(F_{c,tg}*120)$=

   = $\frac{1}{2}*F_{c,tg}*120/(190*A_b)$

σ_t = tensile stress in bolts A and B due to moment $(F_{c,rad}*65)$

   = $\frac{1}{2}*F_{c,rad}*65/(210*A_b)$

σ_e = $\sqrt{\{\sigma_t^2 + 2*(\tau_1^2+\tau_2^2)\}}$

static load: σ_e= 53 N/mm² ($\bar{\sigma}$= 160 N/mm²)

fatigue load: σ_e= 38 N/mm² ($\bar{\sigma}$= 60 N/mm²)

Conclusions:

The stresses don't overstep the admissible stresses ($\bar{\sigma}$).
Therefore the construction, shown in figure 3, is strong enough.

REMARK :

K 5   K-weld of 5 mm all round

APPENDIX 6	FIG.3		A4
CRANK WITH VARIABLE LENGTH			
	UNIVERSITEIT TWENTE		

get.: F.BRUGHUIS dat: 27-8-

schaal: 1:2,2

niet gearc. maten ± 0.5 mm
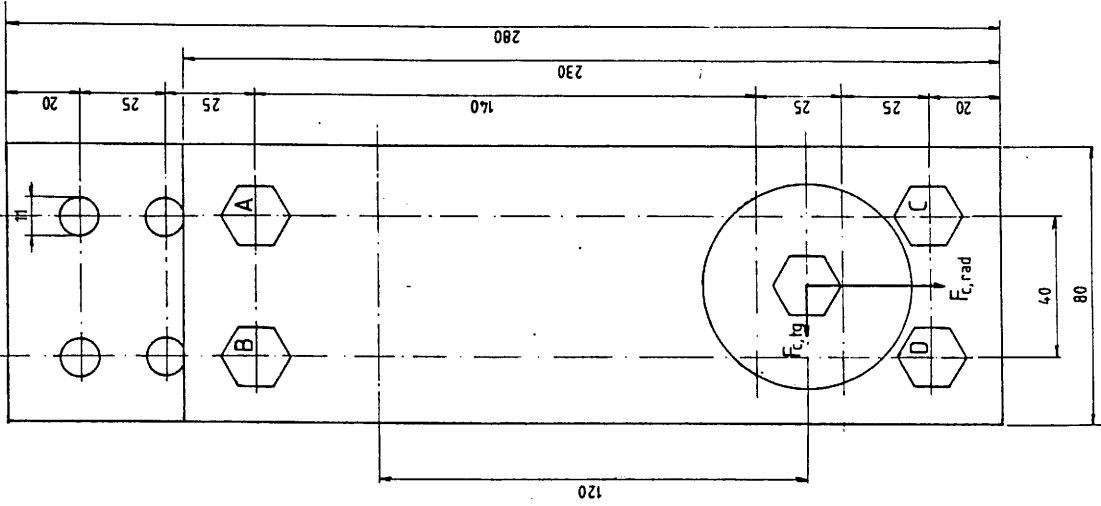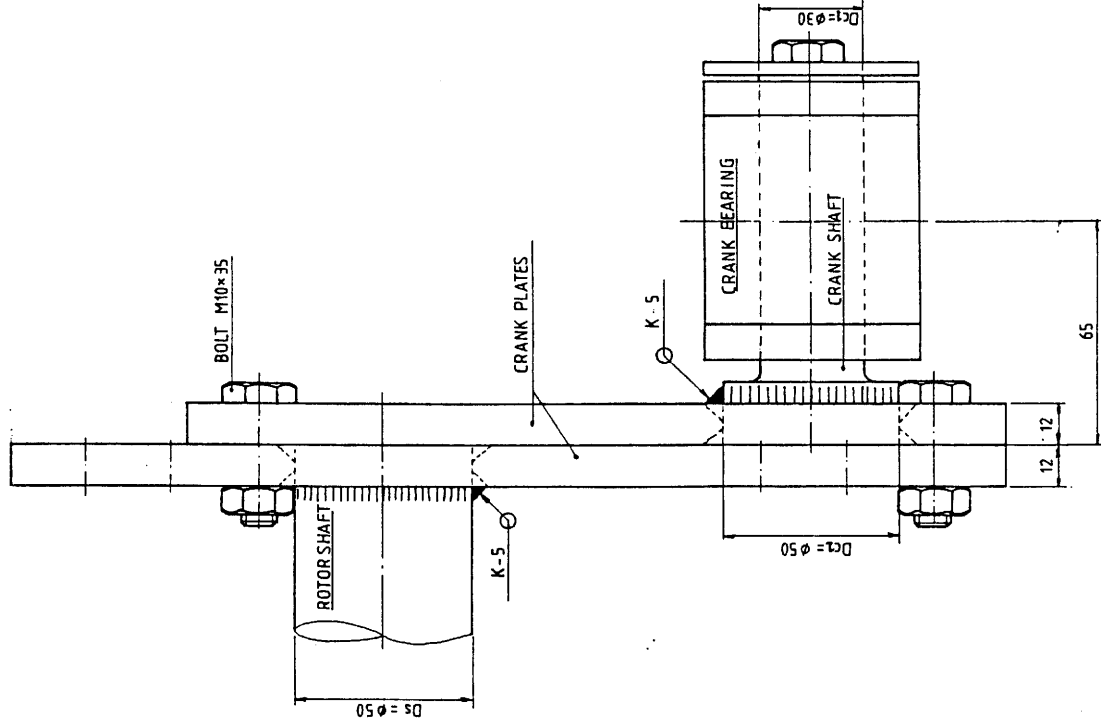
APPENDIX 7: Transmission with ball-bearings

The transmission of drawing nr. 4 has wooden bearings. In some
countries it is easier to obtain ball-bearings than tropical
hard-wood. Then it may be better to use ball-bearings in stead
of wooden bearings, as in figure 1.
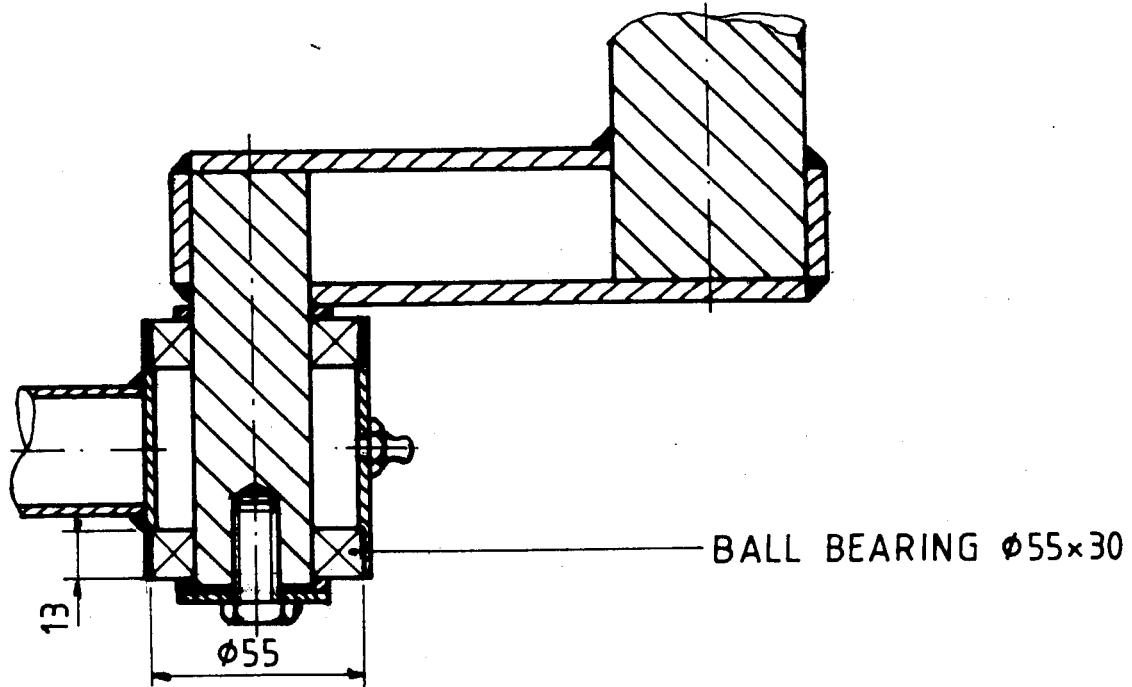
BALL BEARING ⌀55×30

figure 1: cross-section of the crank bearing with ball-bearings
(scale 1:2)

Another construction with ball-bearings is given in figure 2.
This construction has one ball-bearing in stead of two at each
turning-point of the transmission, except the one of the tumbler.
These bearings are self-aligning. Therefore it isn't necessary
that all axises of the turning-points are strict parallel. This
makes the manufacturing much easier.

Ball-bearings can be used for the rotor shaft bearings too
(bearing blocks).

upper view

ROTORSHAFT

CRANK

CONNECTING ROD

BALL-BEARING

TUMBLER

side view

BALL-BEARING

PUMP ROD
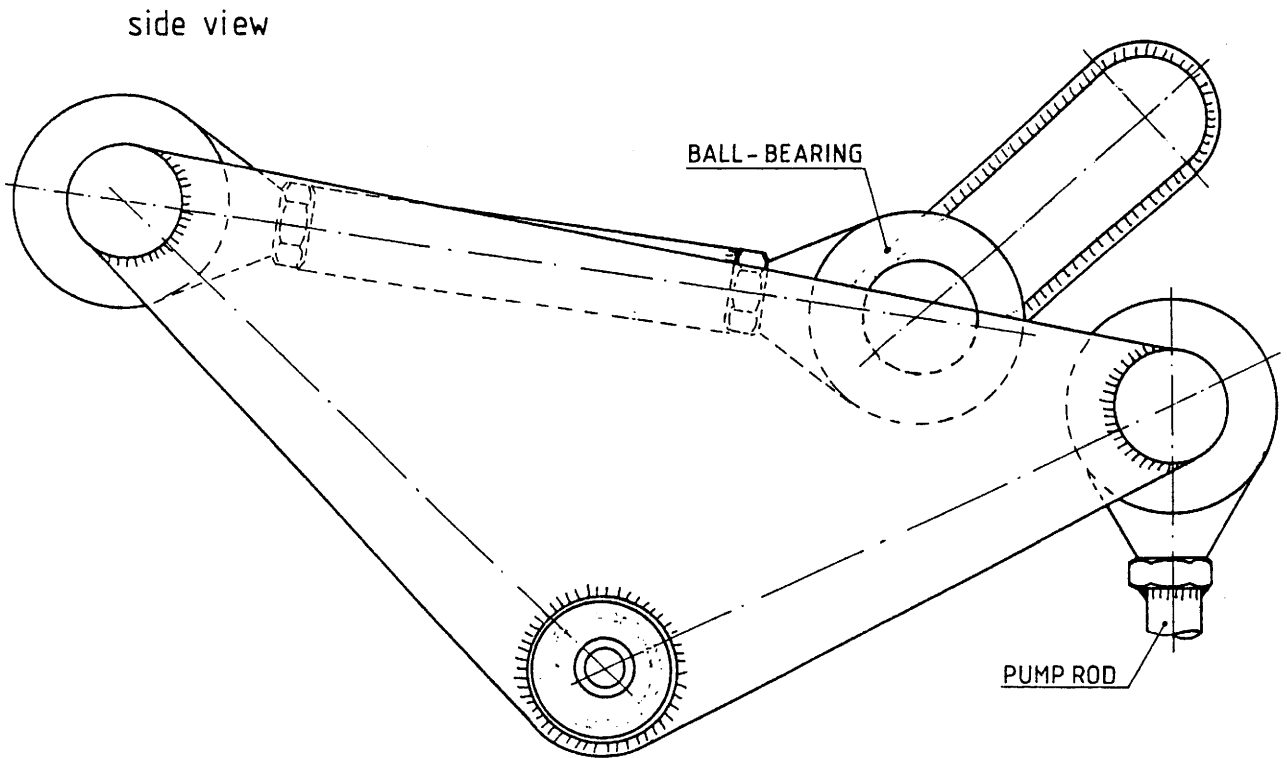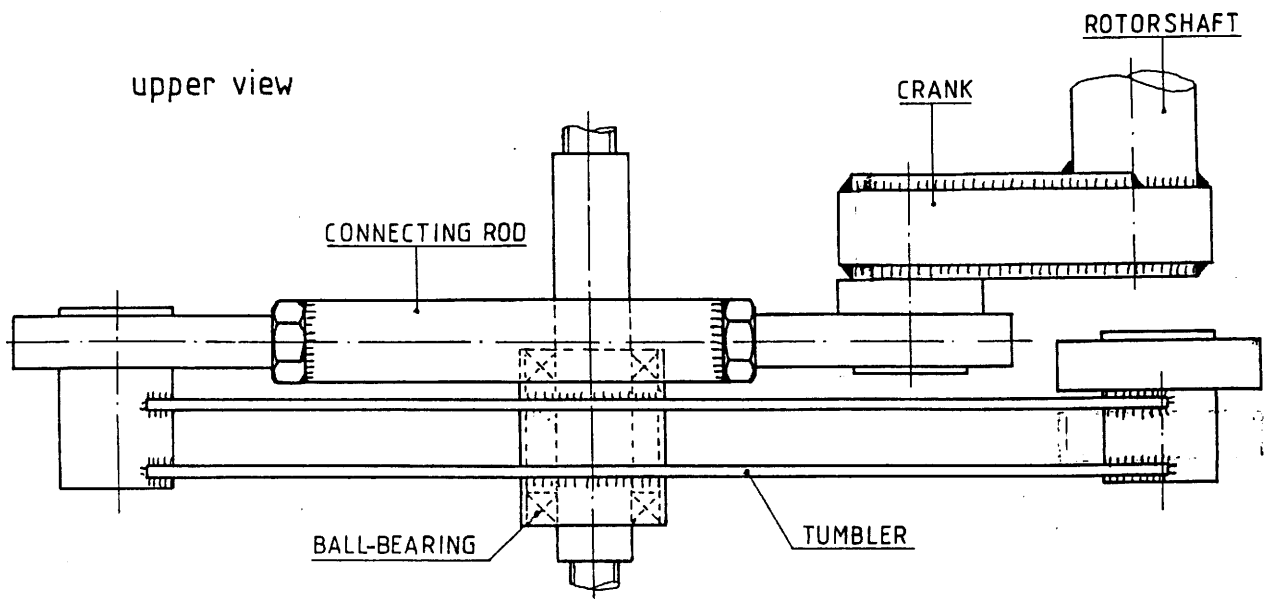
wijziging:	APPENDIX 7	niet getol. maten ± 0,5 M.M.	proj.: ⊕ ⊏⊐ zet.: Brughuis	dat.: 28-6
		schaal: 1:3	gez.:	
	TRANSMISSION DIEVER 450 WITH BALL-BEARINGS			A4
	TECHNISCHE HOGESCHOOL TWENTE			

APPENDIX 8: Installation of the 12PU500

Till so far the following windmill parts are supposed to be
prefabricated: tower, headconstruction, moving parts, rotor
frame and blades, and the pump section. However, this does
not mean that the job has been done. On the contrary: quite
a deal of essential steps are to be undertaken yet.
First of all the parts are composed partly in order to check
the fit. Therefore put the headconstruction on the tower in
order to find out whether it rotates smoothly over the tower
pipe. Also install the moving parts (mainshaft, crosshead and
connecting rod) temporarily and check the functioning. Also
check the fit of some blades (at random). Until and unless
these checks are confirmed the windmill is ready for transport.
One can imagine what inconvenience will be felt on the site if
needed corrections should be carried out over there.
Another important thing with regard to windmill installation
should not be forgotten. Help devices like the jib crane and
lifting hook, which are represented in the drawing, are not
merely helpful but a must! This from a safety point of view.

The use of the jib crane is really necessary for lifting the
head construction and rotor both. If this help device is made
(see drawing) see that the pulling rope runs nicely in a pulley
and cannot run off!
The lifting hook is a help device which provides a suitable at-
tachment to the head construction which is situated above the
point of gravity of the latter. One can imagine what happens
if no lifting hook is applied: the head construction tumbles
over and nobody will be able to handle it during the procedure
of lifting and installing.

If all these required precautions have been taken the windmill
is ready for transport.

The drawing presents three ways of transport of the windmill in
order to reach its destination. See that all the parts are hand-

led with care and tie them up in order to prevent shifts
and mutual movement. It will be evident that for proper
transportation some sloth-flags are applied to get the
other traffic attended of this remarkable transport.

6.1 Erection of the tower

The holes for the foundation are dug at the desired pitch.
Make them of equal depth and wide enough to provide suffi-
cient space for the towerleg anchors and to enable the to-
wer to shift to a certain extent. The tower is laid in
position and a rope is connected to the top portion. A lad-
der (or a set of bamboes) is kept vertically over which
the rope is led. Before pulling the tower upright one has
to be sure whether the ladder (bamboes) as well as the
tower itself cannot fall sidewards. Some extra people can
prevent this by guiding and supporting them.
After the erection procedure the tower is placed vertically.
This is checked by means of a water level gauge which is
kept aside the towerpipe (not at top of the towerpipe edge!)
Some filling material (e.g. bricks) are wedged under the
towerleg angles during the adjustement in vertical sence.
Now the desired position of the centre is checked by means
of a rope and plumb-bob, this with regard to the alignment
of the tower with the centre of the well or the centre of
the tubewell. Again the vertical position is checked and
if both positions are satisfactory the towerlegs are
grouted by means of cement and stone- or brick chips.

Unfortunately for this job
the application of a simple
plumb-bob proves to be some-
times unreliable due to the
influence of the wind. How-
ever by visual means the
centring of the tower can
be checked by looking through
the tower pipe along its
centre-line.

eye

help device
with central
hole

pump rod
guide

to centre
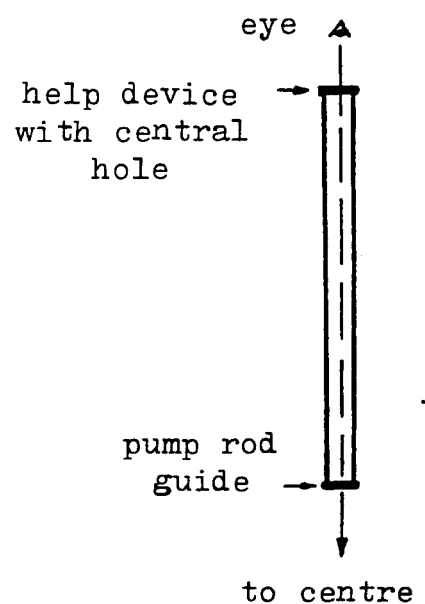
The harding-out procedure of the foundation needs at least
5 days,but in the meantime the time-consuming fixation of
the pump is carried out. As far as grouting is needed (for
pump fixation, washing place etc.) the needed mortar is made
available during grouting the towerlegs.

6.2 Installing the headconstruction

Now the jib crane is going to
prove its worth. It is pulled
and pushed up aside the tower
and finally set on angle 7 of
the tower. See that the jib is
installed at the "windshadow"-
side of the tower, (See drawing)
while its two clamping angle-
iron arms 4 enclose the tower-
pipe in most outward positon.

— ANGLE ⑦ OF TOWER
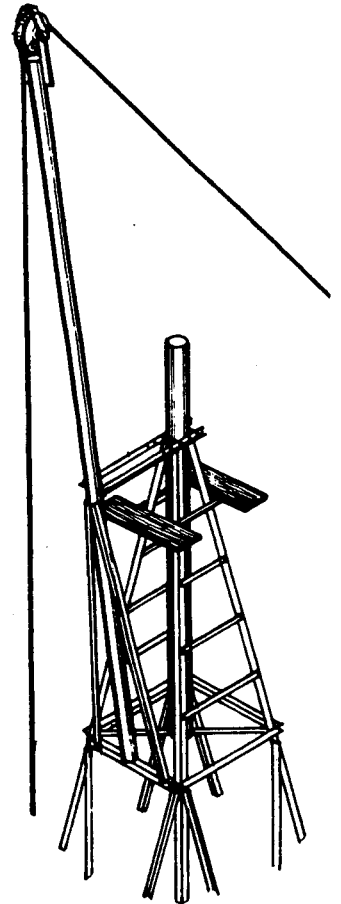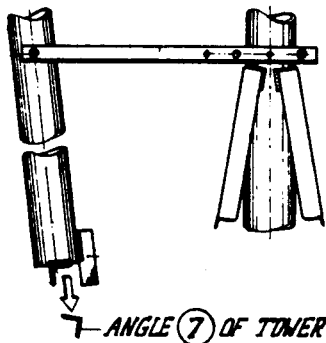
Now the complete headconstruction without help vane is brought
under the jib. The lifting hook is bolted to the tail at about
1.5 metres distance from the tail hinge. The head tends to
turn towards the tail, since the lock mechanism of the help
vane is kept out which makes the entire headconstruction dif-
ficult to handle. Arrange a locking device temporarily either
by roping the upper tail-beam to the headconstruction or by
putting a clamp overthere. Also a pin, stuck through the head
and ending up in the upper tail-beam,serves the purpose very
well. Lock it

The rope is connected to the lifting hook and balancing of the headconstruction is tried out. Balancing in the sense that the front part of the head is out of balance! (say: 5 degrees). This is a <u>must</u> in order to prevent that the framing of the headconstruction, which is going to surround the towerpipe, will grip and throttle it. (This is a most essential precaution which seams to be even contradictional: further on it will be explained).
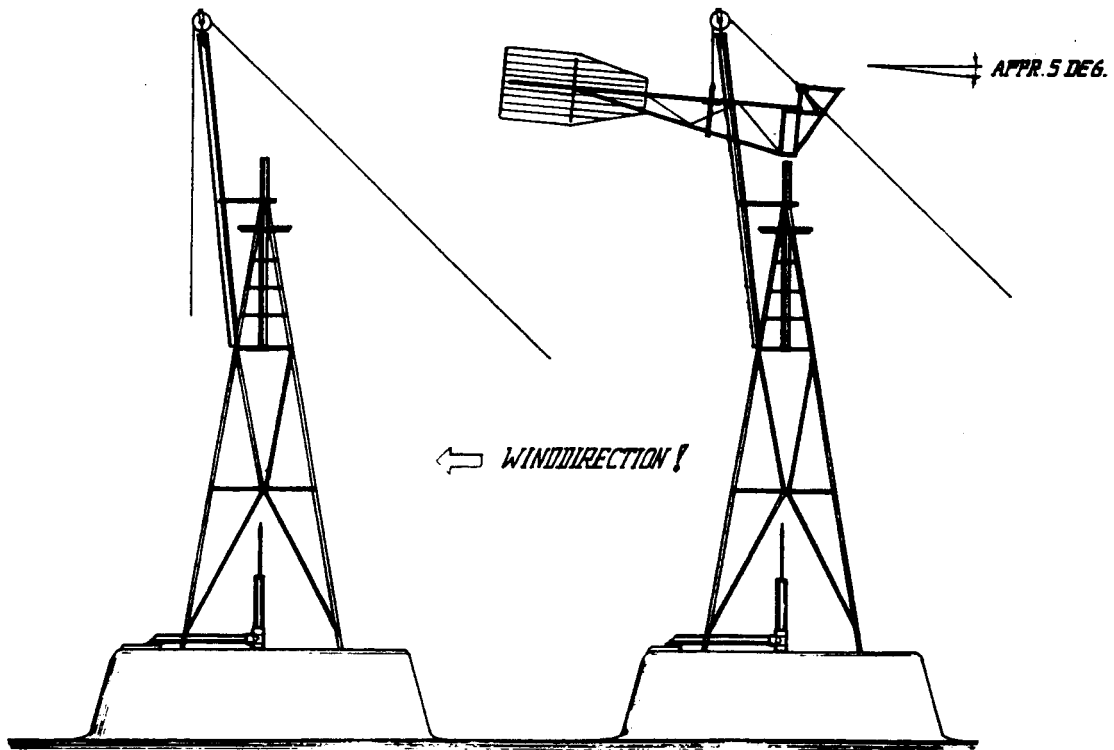
Now the head is lifted slowly by pulling the rope (by approx. 6 persons) and two persons guide the head while climbing the tower. Let the head seek its position in the wind, do not force it and lift it till it floats just above the tower pipe. Both persons, each standing on the platform now, guide the bottom part of the head just over the towerpipe. The head is lowered down very slowly and the persons keep the head pressed down over the towerpipe. Actually they should maintain the "5 degrees out of balance" state of the head while the head itself is lowered down stepwise. The last few centimetres might be bothersome as the inside angles 7 of the head might touch the towerpipe edge and so obstruct a further sliding over of the

head. If extra attention is paid to a proper centration no problems appear and the head sinks into its final position. Loosen the rope and remove the lifting hook and preparations are undertaken now to install the rotor frame.

6.3 Installing the Rotor

First of all the position of the jib crane should be adjusted. Therefore the jib crane is put in the most upright position. Now the headconstruction is turned 180 degrees so that the front part of the head is touching the jib crane. For security reasons we make a rope connection overthere so that the head on which the rotor, shaft and bearing blocks are going to be bolted, should remain under the pulley of the jib crane. This is really necessary as appearing wind might turn the headconstruction suddenly.

The two rotor halves are connected now and the shaft is put into the hub and of course in its original position (the centre points! see chapter 5). The hub bolts are tightened stepwise as well as securing clamping angle irons. Carefully the rotorframe is carried to the tower where it is put in vertical position, standing on one of its six spokes right under the jib crane. Be aware that the extruding shaft points into the lattice of the tower construction and problems will arise during the lifting procedure.

The rope is lowered down now from the jib crane and it is connected to the hub of the rotor frame, but so that the rope passes both the inside – and outside – ring from the back of the rotor!
Carefully and slowly the rotor is lifted and two persons guide it while climbing the tower. Extra attention is paid to the extruding shaft which tends to stick into the lattice of the tower. Finally reaching the level of the platform it is seen that the shaft is positioned on the correct side of the jib crane, since the latter itself will be an obstacle.
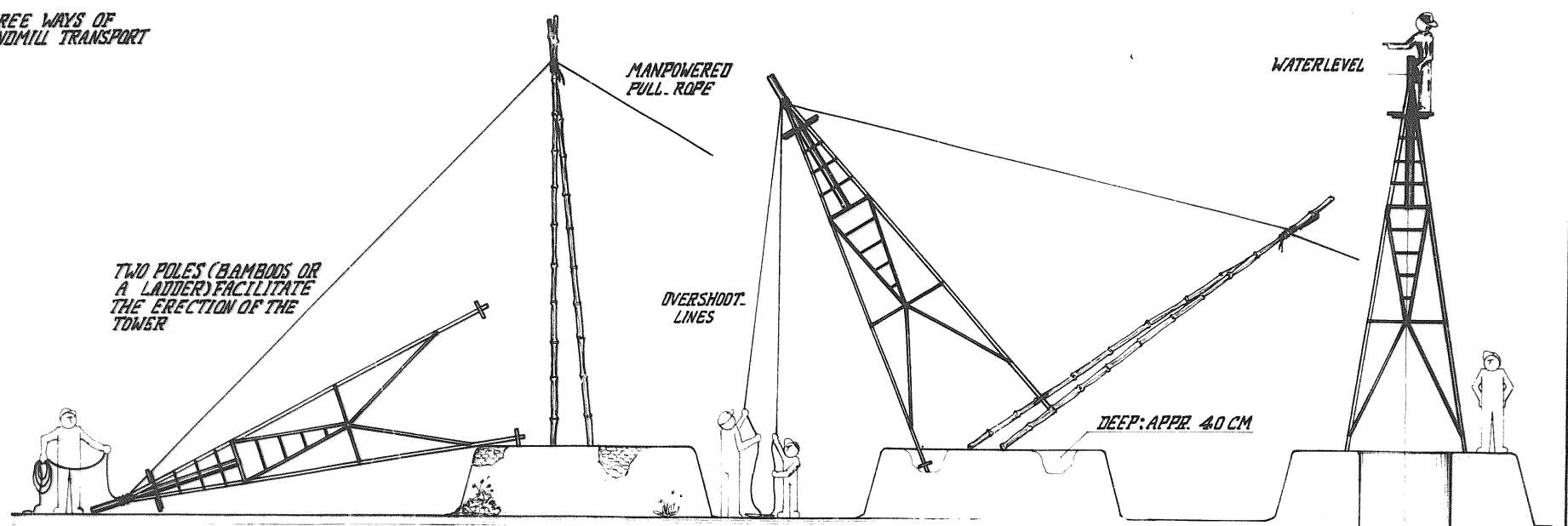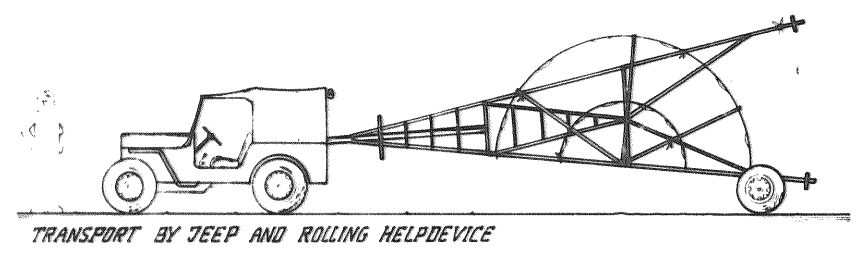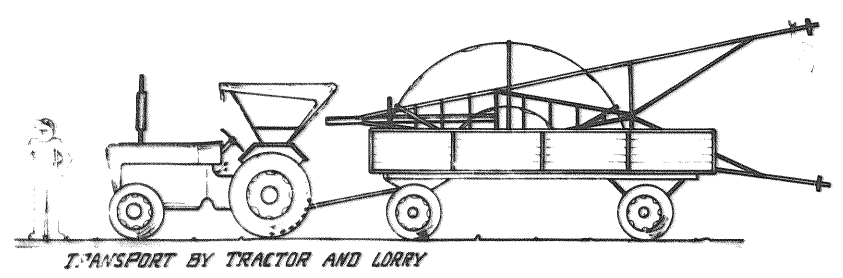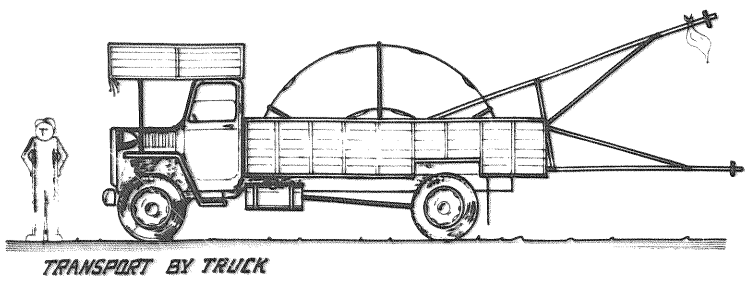
The entire rotor and shaft are hovering over the head now.
With some effort the bearing blocks are positioned above their
supports and the rotor is lowered very slowly then. Quickly
the bolts are put and nuts are screwed by hand first. The
position is secured now and the rope is removed from the rotor
and the jib crane is lowered down. Revolve the rotor during
some revolutions in order to enable the bearing blocks to set-
tle on their supports and tighten the bolts of them finally.

Put the windmill in secured position and bolting of the blades
can start. Therefore one person sits on the platform and an-
other stands on angle 7 of the tower. Bolts, nuts and the
tools can be kept conveniently in a basket which is hung tem-
porarily to the platform during this intensive bolting acti-
vity. It will be evident that the blades are mounted in cross
wise sequence e.g. 1-7, 8-2, 3-9, 10-4 etc.

After this the crosshead, assembled with both connectionrod and
pump rod now, is slided into the towerpipe and the crank bush
and central bolt are assembled. See that locking bolt of the
P.V.C. - towerpipe insert is removed temporarily in order to
allow the crosshead to pass by. Also important is to install
the crank bush with its collar facing the crank plane and do
not forget to apply the washer in between its bolt head and
the bush edge. The crank bolt is tightened up strongly and of
course the locking screw for the P.V.C. - towerpipe insert is
re-installed (and should not be forgotten!). Also the wooden
pump rod guide is installed on bottom side of the towerpipe and
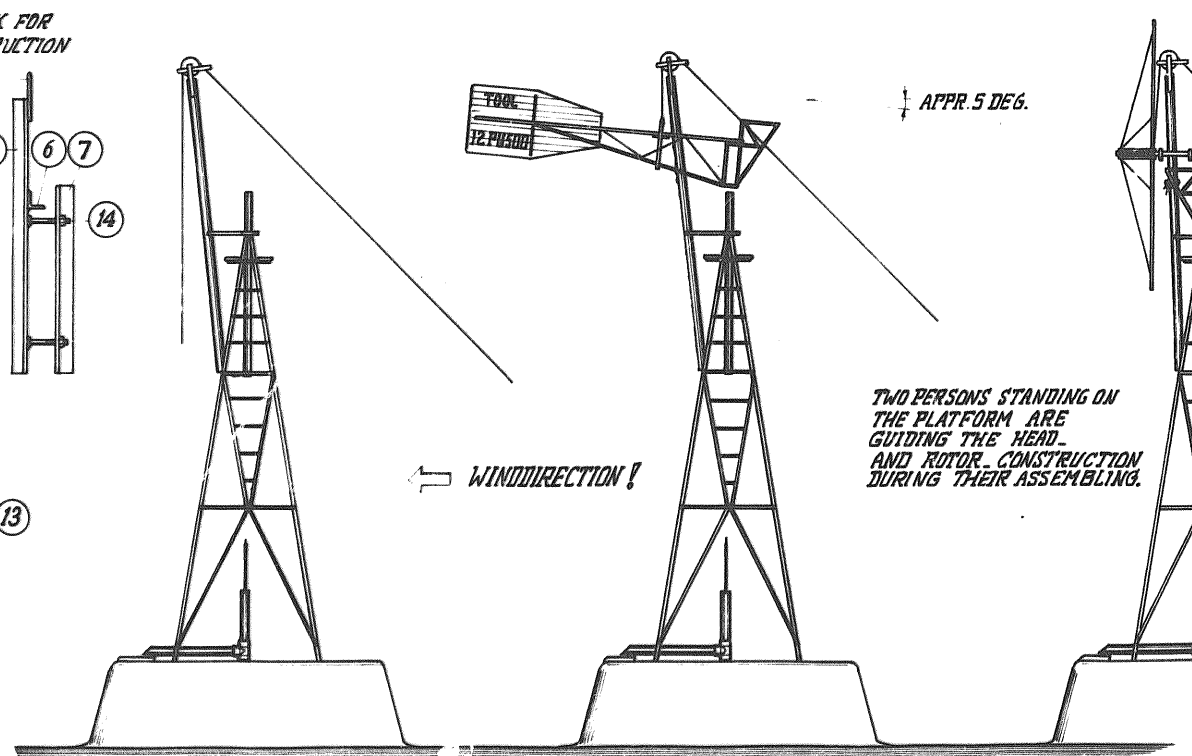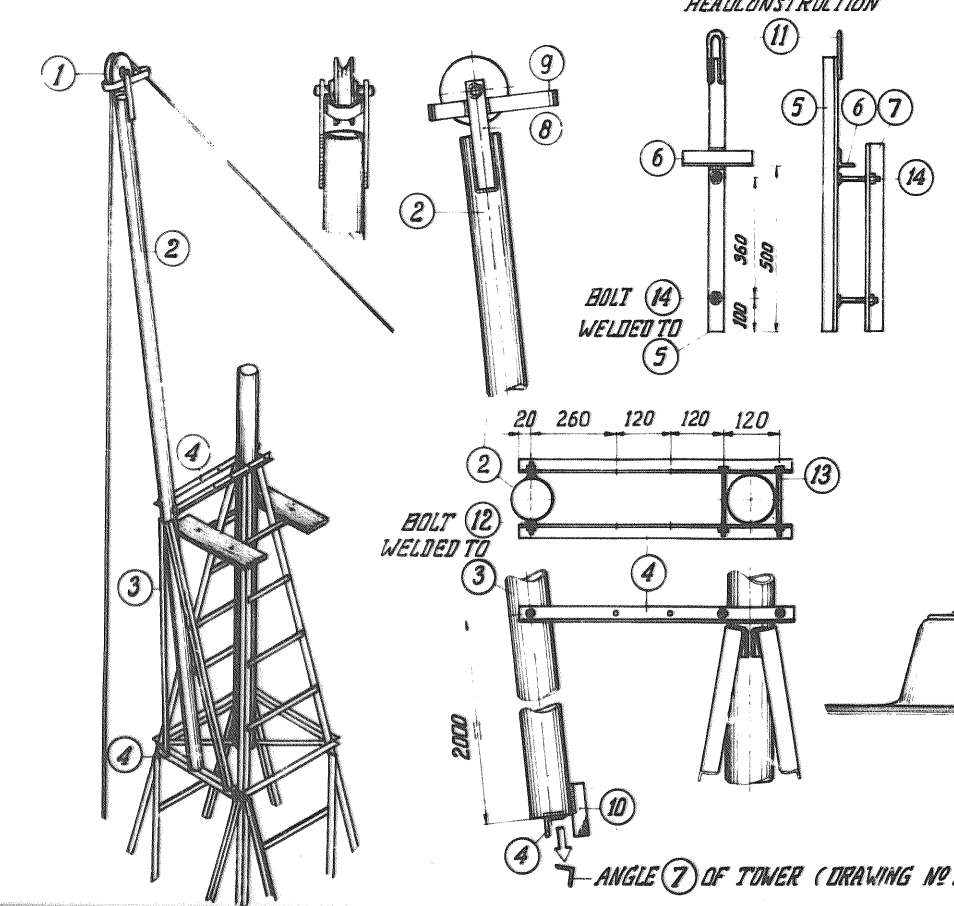locked with the bolt.

Now the helpvane arrangement is built in and the windmill can
be put into the wind enjoying the wind for the first time.

THREE WAYS OF WINDMILL TRANSPORT

TRANSPORT BY TRUCK

TRANSPORT BY TRACTOR AND LORRY

TRANSPORT BY JEEP AND ROLLING HELPDEVICE

MANPOWERED PULL-ROPE

TWO POLES (BAMBOOS OR A LADDER) FACILITATE THE ERECTION OF THE TOWER

OVERSHOOT-LINES

WATERLEVEL

DEEP: APPR. 40 CM

PLUM BOB

IMPORTANT: AFTER PLACING THE TOWER THE TOWERLEGS ARE POORED IN WITH CONGRETE AND AFTER APPR. 5 DAYS THE HEAD- AND ROTOR- CONSTRUCTION CAN BE INSTALLED SAFE AND QUICK BY MEANS OF LIFTING DEVICES. DURING THE DRYING PERIOD OF THE CONCRETE-FUNDATION THE PISTON-PUMP IS FIXED INTO THE WELL ACCORDING TO THE SPECIFICATIONS!

'718'- LIFTING HELPDEVICE

LIFTING HOOK FOR HEADCONSTRUCTION

BOLT ⑭ WELDED TO ⑤

BOLT ⑫ WELDED TO ③

20 260 120 120 120

2000

ANGLE ⑦ OF TOWER (DRAWING № 2)

APPR. 5 DEG.

HEAD IS TEMPORARILY FITTED TO '718' (WIND!)

TOOL 12 PU 500

TOOL 12 PU 500

WINDDIRECTION!

TWO PERSONS STANDING ON THE PLATFORM ARE GUIDING THE HEAD- AND ROTOR- CONSTRUCTION DURING THEIR ASSEMBLING.

SINCE THE CENTRE OF GRAVITY OF THE HEAD-CONSTRUCTION IS SITUATED ABOVE THE UPPER TAIL-ANGLE A SPECIAL LIFTINGHOOK IS APPLIED. A SMOOTH SLIDING OVER THE TOWERPIPE IS ACHIEVED IF THE FRONT IS OUT OF BALANCE APPR. 5 DEGR.

ROTOR IS LIFTED WITH THE SHAFT COMPLETE WITH BEARINGS FIXED IN THE ROTORHUB IN ITS ORIGINAL POSITION (CENTRE POINTS!) AND FIXED ON THE HEAD CONSTRUCTION. NOW THE '718' CAN BE REMOVED AND THE BLADES, SAFETYDEVICE AND MOVING PARTS ARE ASSEMBLED.

№	NUM-BER	MATERIAL/NAME/REMARKS/MEASUREMENTS	
14	2	BOLT NUT WASHER	M 12 x75 (W ½" x 3")
13	2		M 12 x150 (W ½" x 6")
12	2		M 12 x 50 (W ½" x 2")
11	1	ROD	Ø 10 (Ø ⅜") x 200
10	1	FLAT	30 x6 (1¼" x¼") x 100
9	1		x 600
8	2		x 200
7	1	ANGLE IRON	40 x40 x4 (1½"x 1½") x550
6	1		x 200
5	1		x 800
4	1		x 600
3	2		x2000
2	1	GASPIPE	Ø 3" x 4500
1	1	PULLEY-SHAFT AND ROPE OR STEELCABLE (30 M)	
№	NUM-BER	MATERIAL/NAME/REMARKS/MEASUREMENTS	

7 OF 7   INSTALLING THE WINDMILL — 12 PU 500 —

FOR INFORMATION:

Installing the windmill

No	Number required	Materials, particulars and measurements	
1	1	Pulley shaft and rope or steel cable (30 M)	
2	1	Gaspipe	φ3″ × 4500
3	2	Angle iron	40 × 40 × 4 (1½″ × 1½″) × 2000
4	3	,,	,, × 750
5	1	,,	,, × 800
6	1	,,	,, × 200
7	1	,,	,, × 550
8	3	Flat iron	30 × 6 (1¼″ × ¼″) × 200
9	1	,,	,, ≐ tower pipe dia.
10	1	,,	30 × 6 (1¼″ × ¼″) × 100
11	1	Rod	φ10 (φ⅜″) × 200
12	2	Bolt, nut, washer	M 12 × 50 (W ½″ × 2″)
13	1	,,	M 12 × 150 (W ½″ × 6″)
14	2	,,	M 12 × 75 (W ½″ × 3″)

Three methods of transporting the windmill
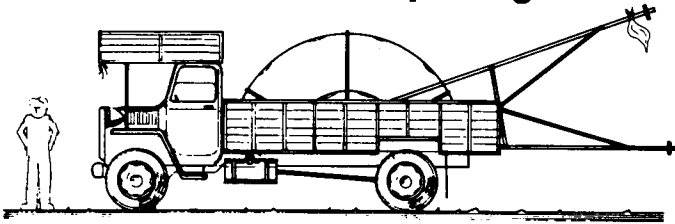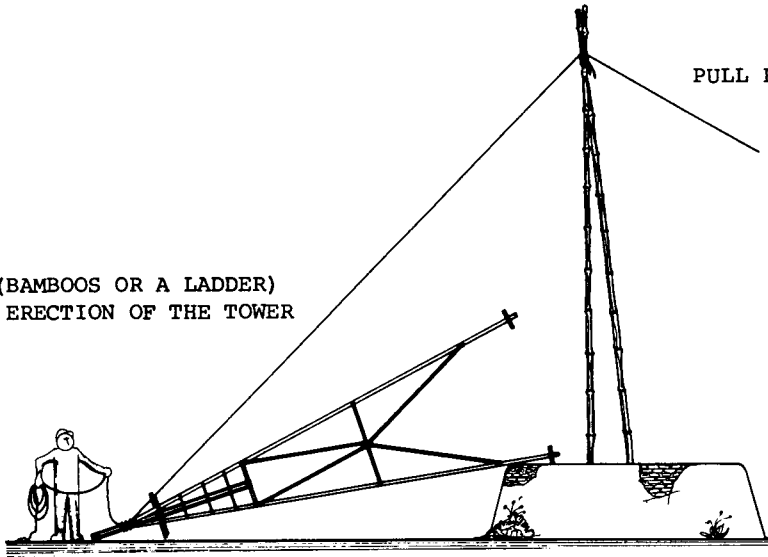

TRANSPORT BY TRUCK
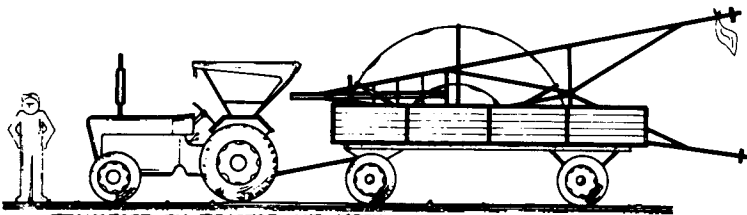
TRANSPORT BY TRACTOR AND LORRY
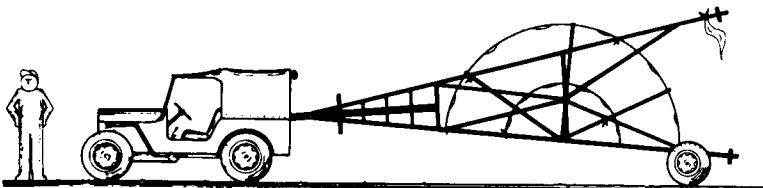
TRANSPORT BY JEEP AND ROLLING HELPDEVICE

PULL ROPE BY MEN

TWO POLES (BAMBOOS OR A LADDER)
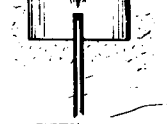FACILITATE ERECTION OF THE TOWER

SPIRIT LEVEL

OVERSHOOT LINES
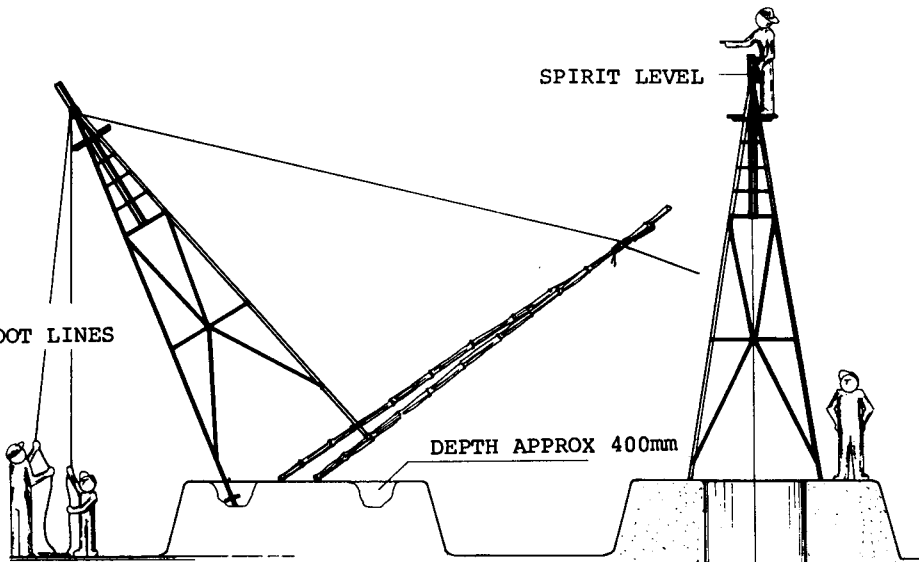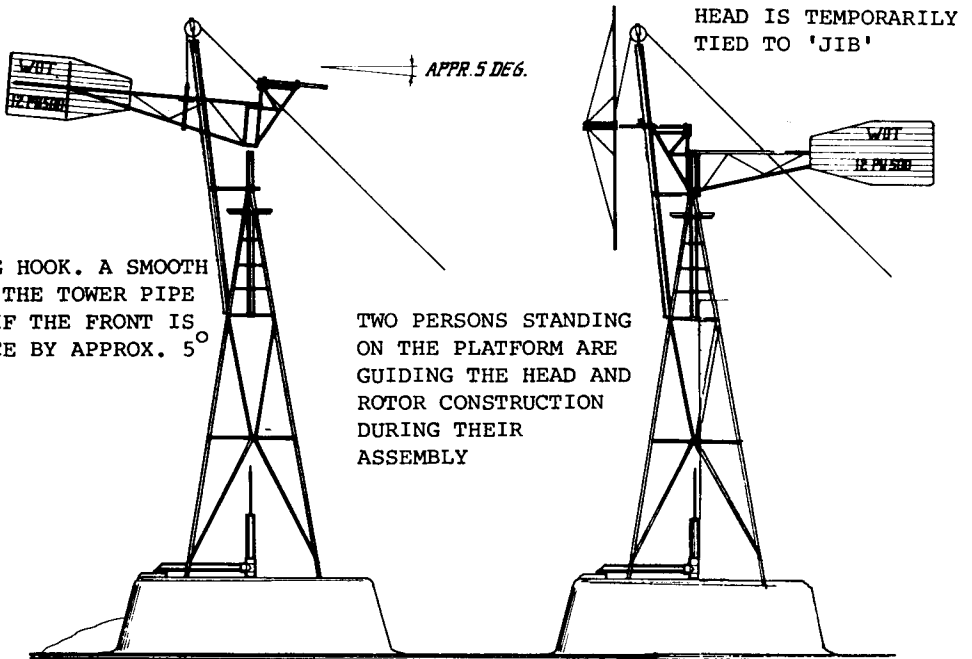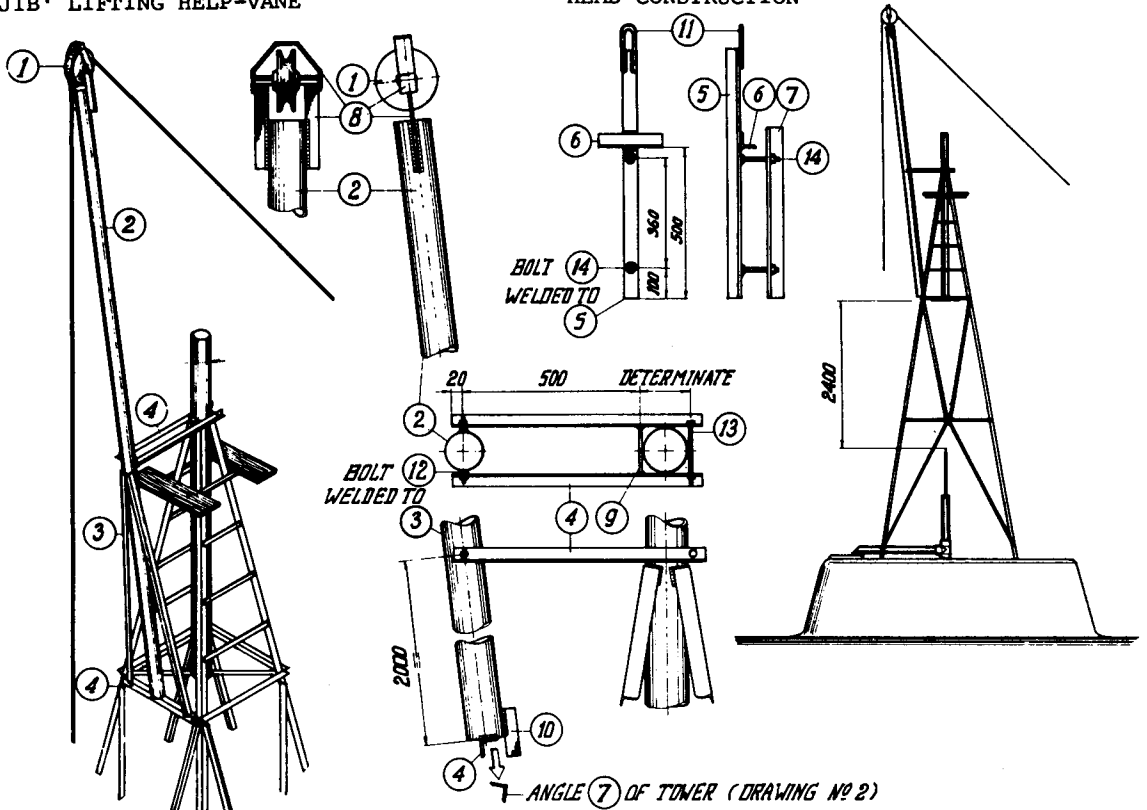
DEPTH APPROX 400mm

PLUMB-BOB

IMPORTANT

AFTER PLACING THE TOWER CONCRETE IS POURED AROUND
THE TOWER LEGS AND AFTER ABOUT 5 DAYS THE HEAD AND
ROTOR CAN BE INSTALLED SAFELY AND QUICKLY BY MEANS
OF LIFTING DEVICES.   DURING THE CURING PERIOD OF
CONCRETE FOUNDATION FOR THE PISTON PUMP CAN BE
FIXED INTO THE WELL AS SPECIFIED.

'JIB' LIFTING HELP-VANE

LIFTING HOOK FOR
HEAD CONSTRUCTION

BOLT ⑭
WELDED TO
⑤

BOLT ⑫
WELDED TO

20   500   DETERMINATE

2000

4 └ ANGLE ⑦ OF TOWER (DRAWING № 2)

2400

APPR. 5 DEG.

W3T
12 PV 500

HEAD IS TEMPORARILY
TIED TO 'JIB'

W3T
12 PV 500

USE A LIFTING HOOK. A SMOOTH
SLIDING OVER THE TOWER PIPE
IS ACHIEVED IF THE FRONT IS
OUT OF BALANCE BY APPROX. 5°

TWO PERSONS STANDING
ON THE PLATFORM ARE
GUIDING THE HEAD AND
ROTOR CONSTRUCTION
DURING THEIR
ASSEMBLY

ROTOR IS LIFTED AND POSITION AT THE TOP OF ITS SHAFT AND
SHIFTED IN ITS ORIGINAL POSITION AND FIXED. REMOVE 'JIB'
AND ASSEMBLE BLADES, SAFETY DEVICE AND MOVING PARTS.