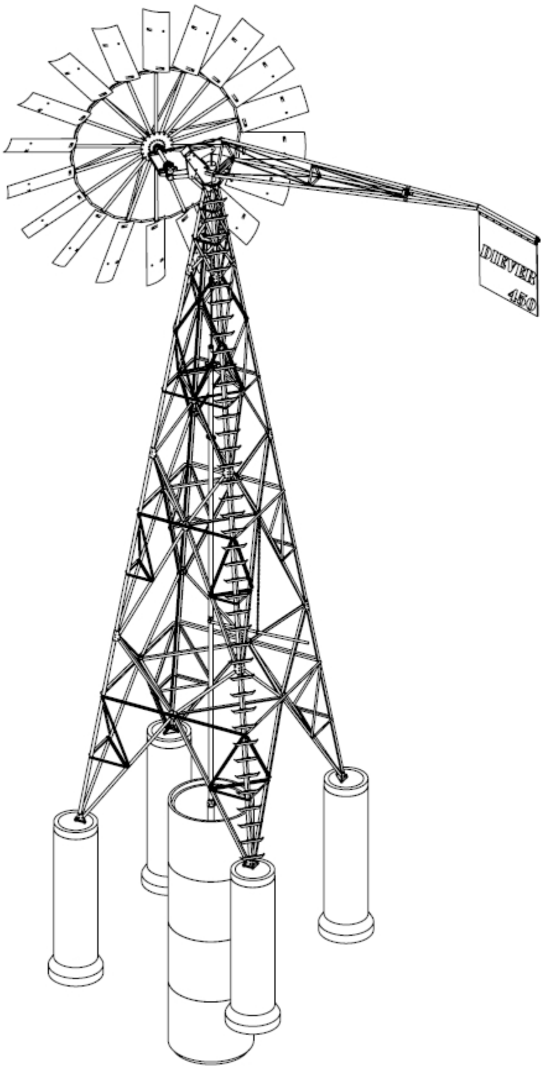
TECHNICAL REPORT 2018 DIEVER 450



Working Group on Development Techniques publication

TECHNICAL REPORT 2018 DIEVER 450

windpump

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Foreword

Content of this report is an addition to notes which appear on fabrication drawings of a slow running (metal) windmill in combination with a reciprocating pump. The design has an intermediate technological character. That is to say the aim is/was to come up with a design which can be entirely constructed from materials which are nowadays available worldwide and moreover, can be re-used after its lifespan has been exceeded. Principle of designing cradle to cradle. Materials used are (stainless) steel, brass and some plastics. Connections have been made using fasteners M5, M6, M8, M10, M12 and M16 (all class 8.8). For bearings it is chosen to use commercially obtainable rolling bearings. Workforce, for manufacturing and installing of this windpump, should be reasonably educated/experienced in mechanical engineering and able to interprete technical drawings correct. A workshop containing a lathe, drilling- and milling machine, welding equipment etc. is necessary. For safe installation a 1000kg chain hoist, hoist slings and personal safety devices (all certified) should be provided. Renting a mobile crane might be more cost effective (when regarding time as money).

Fig. 0.1: installing of pump (photo 2016)

Contents

Fig. 0.2: hoisting of rotor (photo 2005)

1 History

The Diever 450 is a water-pumping windmill designed by WOT in 1987 as a successor of the 12PU500 (also designed by WOT in 1979). The Diever 450, in fact 18PU450, is named after a prominent WOT member, Albert (Appie) Diever (a former light-weight boxer). A prototype of (t)his mill was erected at testfield of WOT in 1987. In 1990 a Technical Report of Diever 450 [lit. 4] was issued and new drawings [lit. 3] were made. The former safety mechanism [lit. 1] was changed to an inclined hinge main vane with sidevane. However, the prototype on the testfield was not changed. Instead, in 1991, approximately 10 Dievers were installed in Bolivia according to these new drawings as a tryout. The feedback-report of this tryout, which was received in 1992 was positive [lit. 7]. In 1993 the windmill on WOT testfield was equipped with a hinged side vane with an eccentric rotor (according to Kragten Design). For this reason the head, the tail with vane and the transmission had to be altered [lit. 5 & lit. 6]. The complete structure was repainted and new wooden bearings were fitted. Painting job was redone in 2004/2005. Several graduates were busy with finishing drawings of rotor [lit. 10], tail and tower [lit. 11] and a study about optimising the transmission [lit. 9] was carried out. Parallel to this a thorough (comparing) survey was made to verify the applicability of Diever 450 regarding technical, producible and commercial aspects [lit. 12]. In 2010 Diever 450 was built by Pulchowk Engineering Campus, Nepal (still according to drawings of lit. 3). In 2016 Diever 450 on testfield needed another layer of paint. It was decided to equip the mill with rolling bearings, according to drawings from 2004, main reasons being reducing friction and maintenance. In 2017 fabrication drawings [lit. 13] of Diever 450 were completed. During spring 2018 permission was granted to fabricate one Diever 450 (according to lit. 13) in South Africa. That decision triggered WOT to issue a "new" technical report of Diever 450, with all changes made since 1990. When compiling and checking data, several improvements were suggested to fabricator in South Africa. It is expected, a feed-back report will enlighten WOT of effectiveness of proposed improvements [lit. 15].

Author thanks all who served developing Diever 450 and/or maintaining the prototype of Diever 450 on testfield for 30+ years! You just are with too many to name you all. Keep it up!

Fig. 1.1: Albert Diever (photo 1987)

Fig. 1.2: first Diever 450 (photo 1987)

Bolivia (photos 1992)

Nepal (photo 2010)

Netherlands (photo 2018)

South Africa (photo 2018)

Fig. 1.3: Diever 450 locations (known to WOT)

2 General

Diever 450 is a slow-running water-pumping windmill (λ_d = 1 at design windspeed v_d = 3 m/s) with a rotordiameter of 4,5 meter hosting 18 (metal) blades, each with a length of 1 meter and 400mm wide. Hubheight is approx. 11 meters. Other hub heights may be chosen. The tower has four legs. The windmill has an automatic safety system which turns the rotor gradually out of the wind with increasing windspeed and turns the rotor back into the wind when windspeed drops. Survival windspeed of Diever 450 is set on 40m/s. Above that, damage to the windmill may occur.

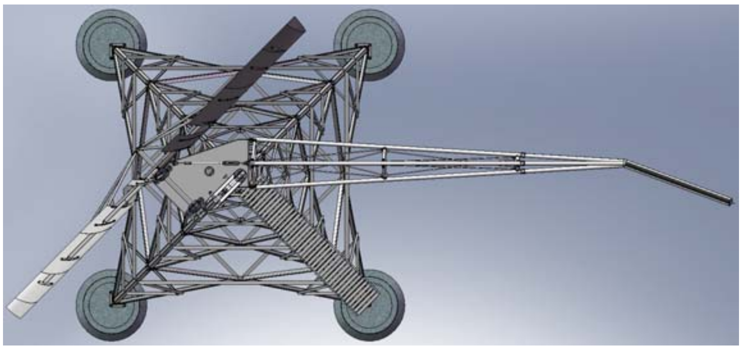
Parts of Diever 450 were checked for different load cases:
- Fatigue: rated windspeed v_r = 9 m/s, rotor 60rpm ($\lambda_r \approx 1,6$), rotor facing wind.
- Static: operating windspeed v_o = 12 m/s, rotor 80rpm ($\lambda_o \approx 1,6$), rotor facing wind.
- Static: survival windspeed v_s = 40 m/s, rotor 80rpm, rotor out of wind ($\delta \approx 70^o$).

The transmission uses a four bar mechanism to facilitate the piston to take a longer time moving upward than the time it takes to move downward.
Incorporated in the pumprod are a load limiter to disconnect the transmission from the pump in case of pump failure, and a shockabsorber to smoothen peakforces.
The pump consists of a stainless steel cylinder and its inner parts are made of brass to prevent corrosion.
Preferably Diever 450 should not be installed near trees or buildings as turbulence round these objects influences proper functioning of the windmill [lit. 12].
Diever 450 needs to be placed straight above a borehole with a minimum inner diameter of 4” (approx. Ø 100mm) or above a well, both with sufficient water capacity, particularly in the dry season.
As the average (annual) windspeed is a very important factor for the output of any windmill, a location with a sufficient wind regime is required for successful performance of Diever 450.

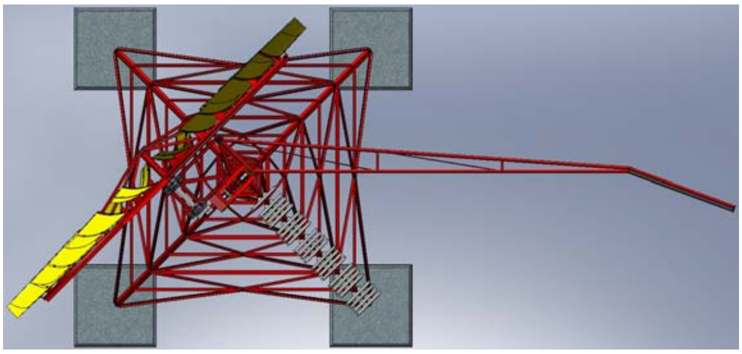
In regard to this report:
- Fabrication methods and choices made, to come to this design are within this report, solely as a bonus to fabrication drawings.
- Only assemblies and pictures are presented. This report is not a substitute for fabrication drawings of Diever 450.
- These can be provided only by WOT, and should NOT be taken from the interweb. Please contact WOT, if you have obtained drawings from another source. In that way we can assure you are using the latest version of drawings.
- Fabrication drawings should speak for themselves; it should be possible to manufacture parts without knowing its function. But it proves to be handy most of the time. That is the main purpose of this report.
- Yield, costs, lifespan etc. are NOT in scope with this report. If one is interested in one or more of these topics, see lit. 12 and lit. 15.
- As WOT experienced, painting is quite expensive (if one has to do it approx. every 10 years) and very time consuming. To avoid this, a Diever 450, which can be galvanized might be a solution. One should decide whether one wants to paint or to galvanize, once chosen, one should stick to it for the complete structure as some parts are not interchangeable.

- This report proposes a couple of changes to improve reliability and output of Diever 450, leading to different versions. These versions have been numbered according:

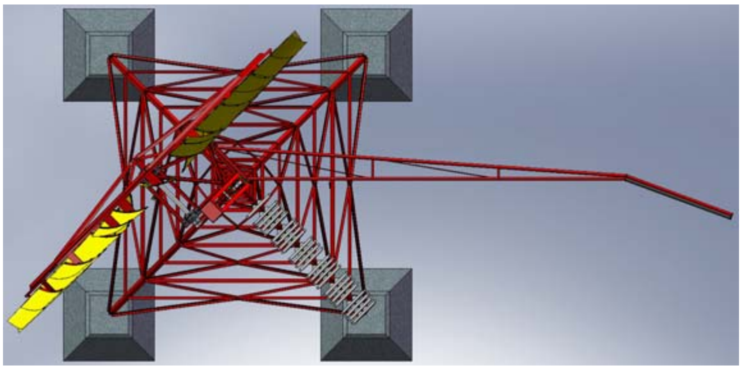
DA:
galvanized
(not yet issued)



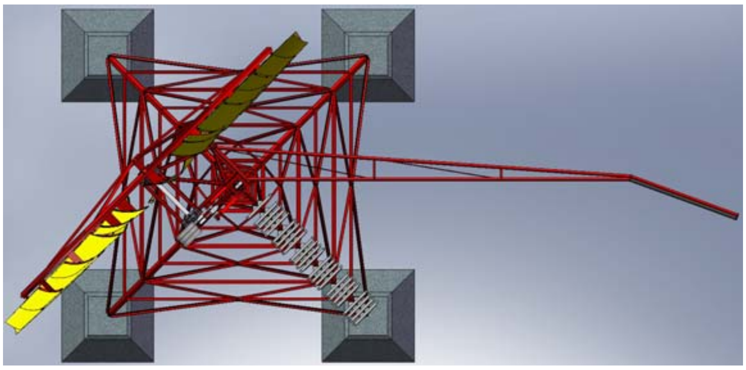
DB:
painted, as on WOT testfield



DC:
as DB, with rotor improvements



DD:
as DC, with possibility of balancing weight of pumprods



It is intended to limit versions to two only (one galvanized and one painted). Experiences from fabricator in South Africa (who is building a Diever 450 according to DD) together with performance of mill when installed, will, most likely, provide useful information which versions are to be preferred.

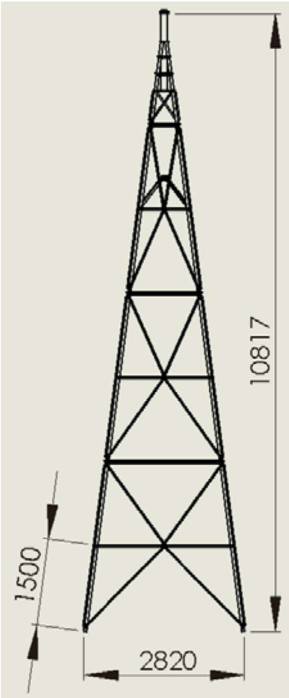
3 Tower

3.1 General

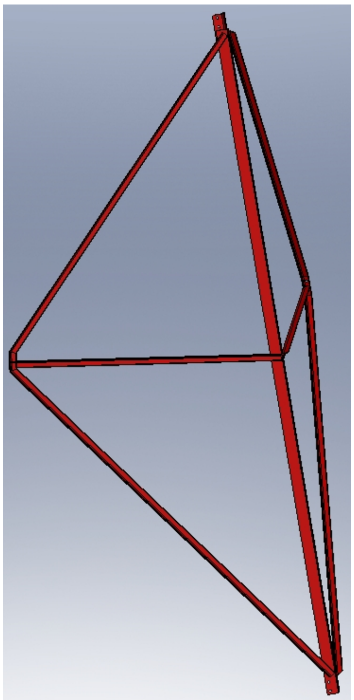
In 1990 (semi)final drawings of the tower [lit. 3]. were made after analytic strength calculations showed that the original tower of the prototype (though it is still standing on the testfield of WOT today) was structurally to weak. In 1994 the material stresses in the tower were re-checked with a computer using the finite element method (FEM) [lit. 8]. As a result, some recommendations, to improve the strength of the tower, were made. These were processed in both (painted and galvanized) versions of which drawings were made in 2017 [lit. 13].

3.2 Painted tower

Weight: approx. 380kg. Legs ∟50.50.5, diagonals and most horizontals ∟30.30.3, some horizontals ∟40.40.4. Distance between leg-connections is 1,5m. Max. subsection height: 3 meter. Subsections may be difficult to transport and/or galvanize, due to their 3D structure.



dimensions



welded subsection

section 4 (photo 2005)

Fig 3.1: painted tower

The tower consists of 4 sections. Three sections have a leg-length of 3 meter, each having four welded subsections. The top section (called: section 1) has a leg-length of 1,5 meter and is completely welded with a flamepipe 4” (Ø 114,3 x 6,3).

Horizontal beams can be used as steps. The pipe functions as a base for the yaw bearings which are fitted to the head (the head with yaw bearings can rotate around this pipe). All subsections are fastened with M10. In case work (for example: re-painting) needs to be done to the head of the windmill, it is easiest to lift the head together with section 1 from the rest of the tower and disassemble both on the ground (presuming the tail and the rotor are already lowered before).

section 1 with vertical limiter section 2 with rising main clamp

Fig. 3.2: tower sections

Around the towerpipe of section 1 a vertical limiter is clamped, to prevent lifting the head by unexpected forces in the pumprod. Its position can be adjusted so, that just a little vertical play for the yaw bearings remains. Rising main clamp (designed for 2”, but 2½" will also fit) is situated in section 2. By clamping the rising main it is possible to fine-adjust the depth of the pump in borehole or well. One should realize, once the pumprods are made to length, there will remain little play in adjusting the pump depth.

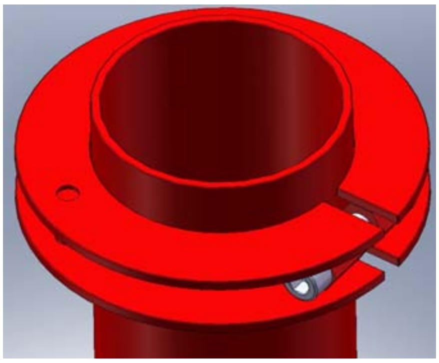


Fig. 3.3: vertical limiter

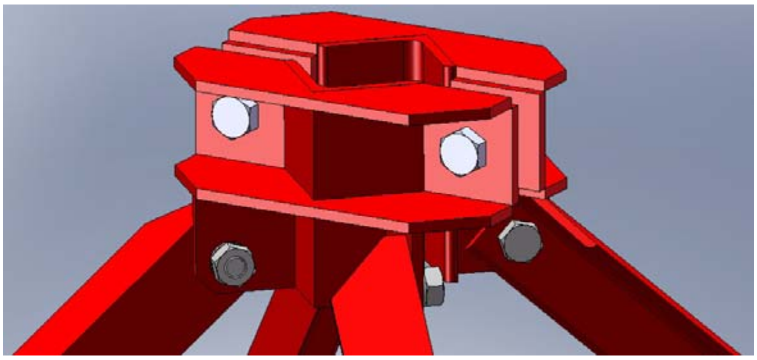
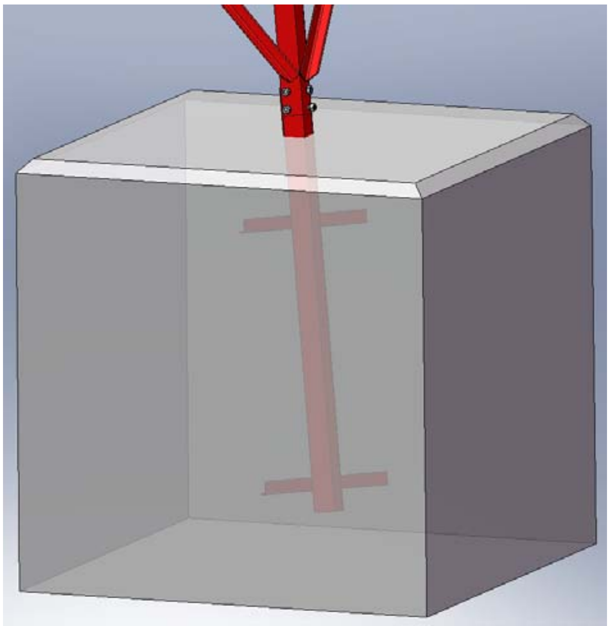


Fig. 3.4: rising main clamp

Base of the tower (section 4) is connected with 4 concrete foundation blocks, which are poured on site. To get the distances and incline of the anchors, which are poured into the concrete, correct, it is advised to erect section 4 and bolt four horizontal beams at the top of section 4, place the structure exactly vertical and at its correct position and connect the anchors to the structure, before pouring the concrete. Instead, a jig might also be useful. For size of concrete blocks and their spacing, see chapters 9.3 and 9.4. No hinges are provided for erecting the complete tower after the concrete has hardened. The tower has to be built up, gradually, by means of a jib or a mobile crane.


anchor and foundation

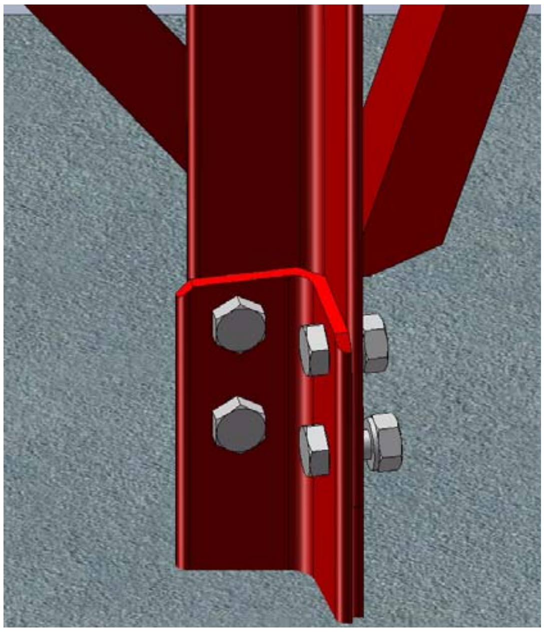
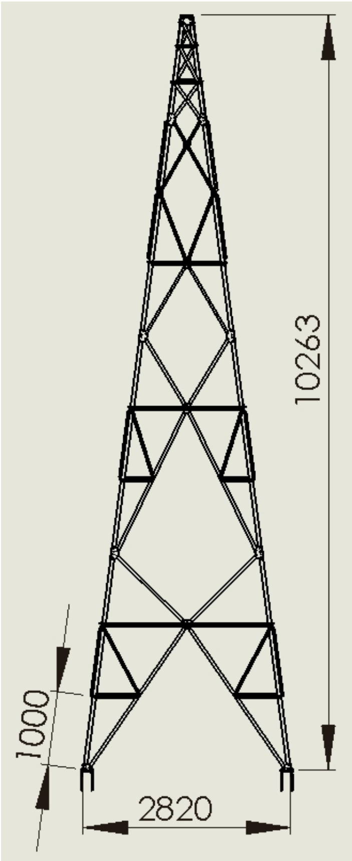

anchor connection

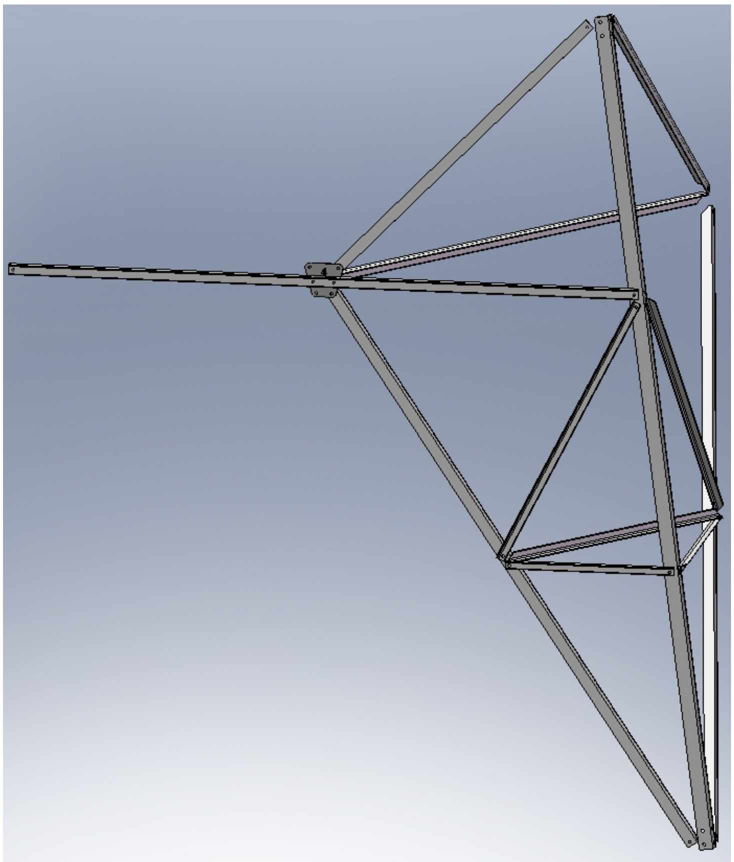
Fig. 3.5: section 4 with foundation

3.3 Galvanized tower

Weight: approx. 480kg. Legs ∟50.50.5, most diagonals and most horizontals ∟30.30.3, some diagonals and some horizontals ∟40.40.4. Horizontal diagonals are added to increase torsion stability. Buckling length is set on 1000mm. Max. length of each part: 3 meter. Parts are easy to transport and/or galvanize. As can be seen, there are large “holes” in each side of the tower. This is done to give passage for rising mains with lengths up to 6 meter to enter the tower. However, it is recommended to limit rising main lengths to 3 meter, because of weight handling.



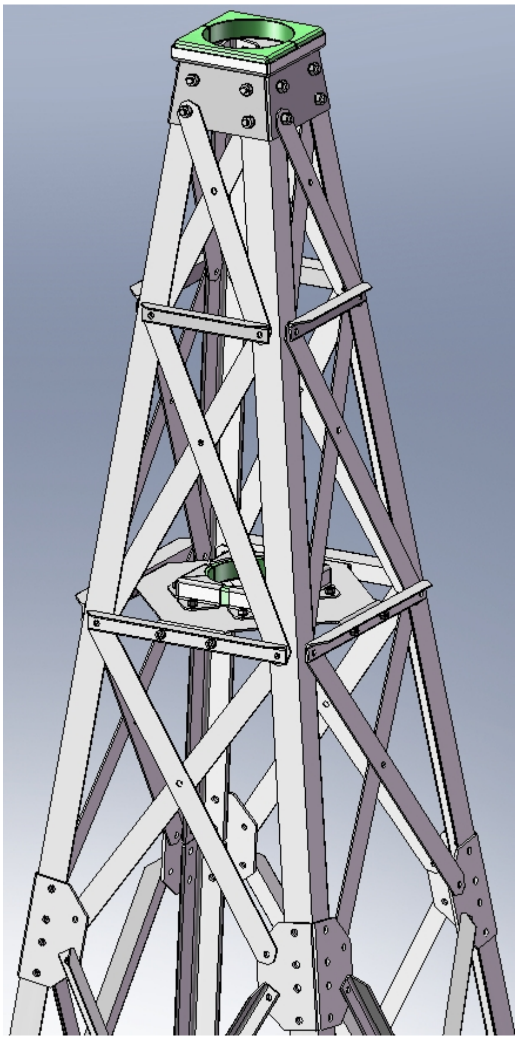
dimensions



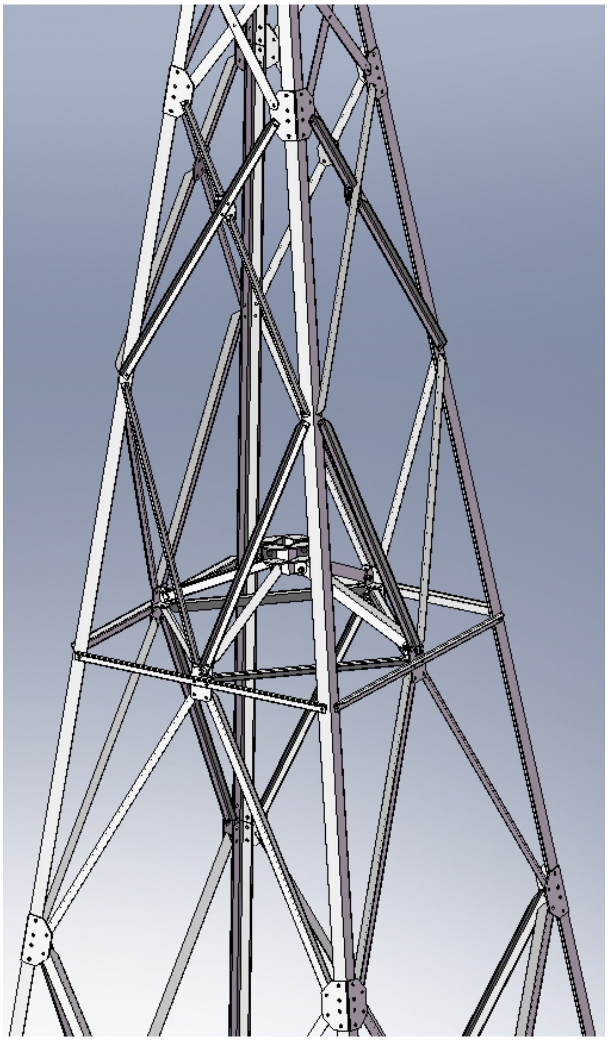
bolted subsection (with two horizontal diagonals)

Fig 3.6: galvanized tower

The tower consists of 4 sections. Three sections have a leg-length of 3 meter, each having four bolted subsections. The top section (called: section 1) has a leg-length of 1,4 meter and it houses two yaw bearings, also bolted. A flame pipe 5” (Ø 139,7 x 4) welded to the head can rotate in these yaw bearings. All subsections are fastened with M10. Parts in subsections are fastened with M8. To ease assembling on site it is advisable to mark each part corresponding its section. The yaw bearings can be replaced without removing the head from the tower. Advantages of this design, compared with the welded structure, is that the hub height of the rotor is lowered significant closer to the top yaw bearing, thus reducing bending moments in the connecting pipe between head and tower and also, the two yaw bearings are better weather-protected. Vertical limiter and rising main clamp have about the same shape as shown in fig. 3.3 and fig. 3.4 and therefore need not to be addressed.



section 1 with yaw bearings



section 2 with rising main clamp

Fig. 3.7: tower sections

Base of the standard tower (section 4) is connected with 4 concrete foundation blocks, which are poured on site. Tower has on each leg a foot, which by means of stud bolts get connected with the foundation. In that manner the tower can be set up and vertically levelled after the concrete has hardened. A simple jig can be used to position the studs on their correct (horizontal) distances.
For situations a mobile crane is not available, feet with hinges can be provided. In that case, (part of) windmill is assembled horizontally and when done, winched up with a wire rope hoist. This requires a stable anchor point for wire rope hoist, which usually can be found in surrounding area. Wire length must be sufficient to cover the distance and a yoke is necessary. More info can be found in chapter 10.
Just for the idea, it is chosen here to use concrete socket and spigot pipes (size 20'' (Ø 500 x 2400)) as a mould in which concrete is poured after positioning the studs. An advantage of these long pipes is that top of concrete pipe can be chosen way above ground level, thus increasing the hub height of the rotor and as a bonus, guaranteed (rain)water drop-off from the foundations to the surrounding. One may want to use a tripod to position these pipes, as they are quite heavy, even unfilled.

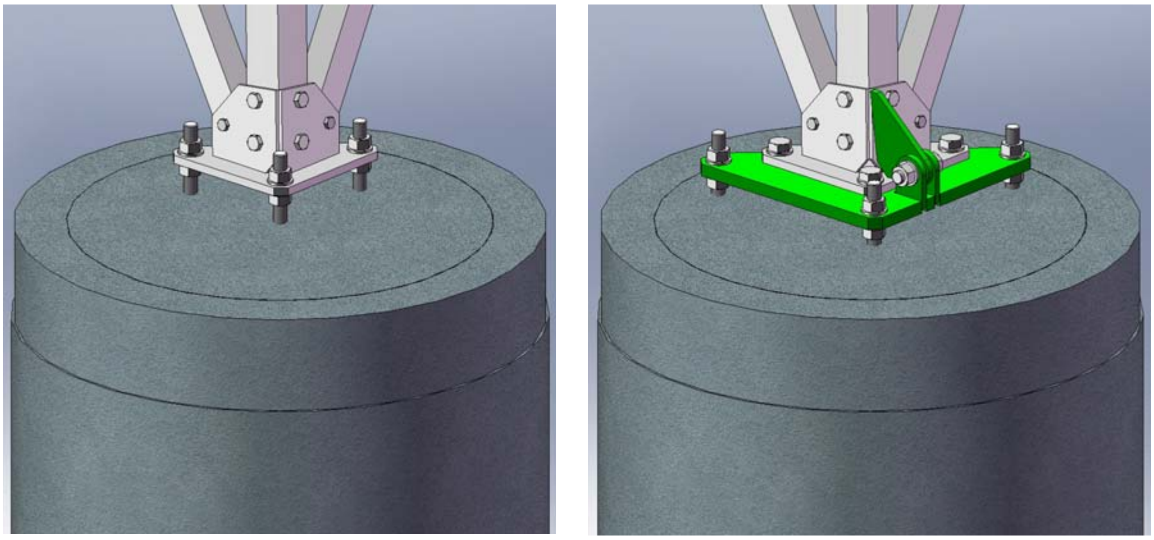


Fig. 3.8: tower foot with foundation

3.4 Extensions

In case obstacles obstruct undisturbed windflow reaching rotor of Diever 450 extensions can be a solution. These will accommodate both towerversions, as they share same base size and incline. Each extension will raise the standard tower by 3 meters and can be galvanized. Three extensions are available, so maximum extension height is 9 meter. All beams are bolted. Approx. weight: section 5, 6 and 7 respectively 204, 232 and 264kg. Realise, tower extensions will impact foundation distances (see chapter 9.4), so think ahead (about growing trees in time).

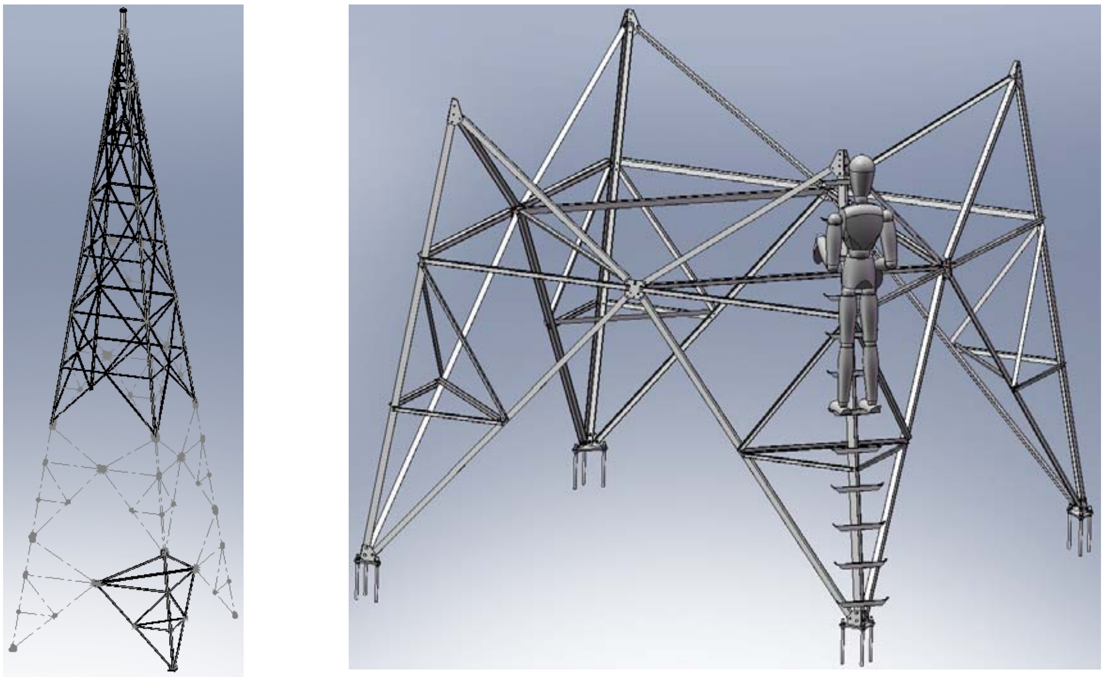


Fig. 3.9: extension (section 6)

3.5 Steps

For regular maintenance of the Diever 450 (lubrication of bearings, preventive changing of breakpin (load limiter) or pump, checking loose bolts etc., once a year) it is necessary to have steps going all the way to the top of the tower. These fit both towerversions. Weight: approx. 0,8kg.

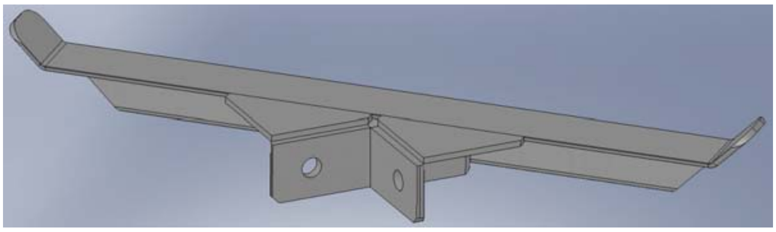
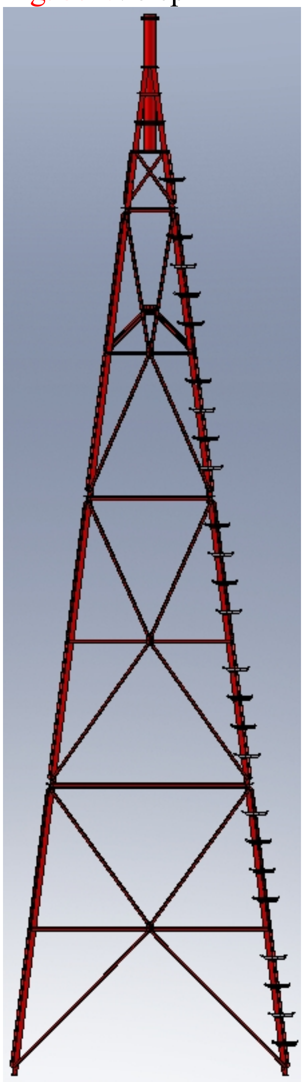
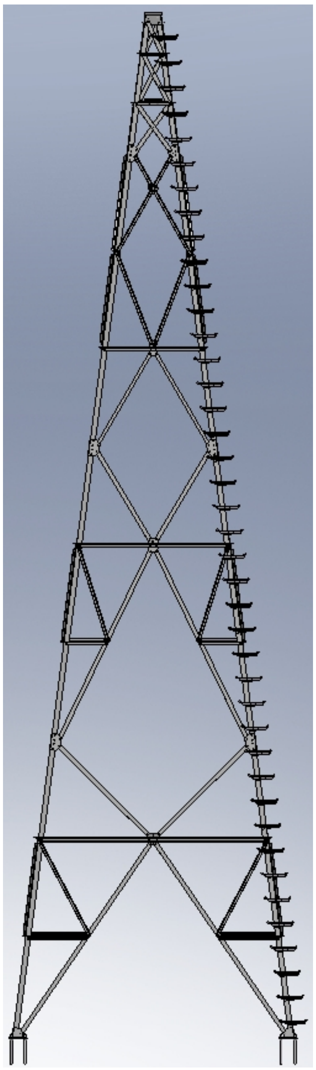


Fig. 3.10: step





painted tower galvanized tower

Fig. 3.11: location of steps

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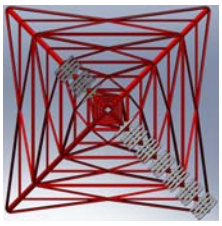


Erect the tower so, that steps are positioned at prevailing lee. In that way tower can be climbed, without getting struck by (rotating) rotor. When there is no prevailing lee, it is advised to mount steps at opposite corners of towersections 1 and 2, in order to obtain safe access at any winddirection.

Fig. 3.12: example of steps at opposite corners of towersection 1 and 2 (top view)

4 Head

4.1 General

The function of the head is to fixate the rotor, transmission and tail of the windmill. It also facilitates the rotor to face the wind by jawing around the centre of the tower. A place for the jib for fitting the rotor and tail is provided. The tail is hinged with regard to the head, to be able to fit the tail by means of the jib. There is a provision to block the rotation of the transmission during maintenance activities. There are two types. The painted head will fit the painted tower, and logically, the galvanized head will fit the galvanized tower. They are NOT interchangeable. In 1992, material stresses in painted head were checked with computer, using finite element method (FEM) [lit. 6].

4.2 Painted head

Weight: approx. 52kg. Construction is mainly made of ∟40.40.4. Many welded connections need chamfering by grinding before welding can take place. Tailhinges are placed under an angle. Because of this, the tail should be fitted before fitting the rotor, dismantling is only possible in reverse order. The axis of the tailvane, which should be horizontal, is not adjustable. Errors in production of the tail will have an impact. Structure houses two yaw bearings, made of HMPE (high molecular polyethylene), also known as Werkstoff-S. It is preferred to use the black version of HMPE, because it is UV-resistant. Both yaw bearings are split to ease mounting.

head, tumbler, conrod, wooden bearings (photo 1993)

head, tumbler, rolling bearings (photo 2016)

Fig. 4.1: painted head

4.3 Galvanized head

Weight: approx. 52kg. Construction is mainly made of sheet metal with a thickness of 3mm, which can be lasercut. DXF files are available. Contains a flamepipe 5” (Ø 139,7 x 4) for two yaw bearings. Top yaw bearing is well protected against rainwater and sunlight. A vertical limiter is clamped at the bottom of the flamepipe, to prevent lifting the head by unexpected forces in the pumprod. At this location there is more elbow room, compared with the welded version, so it is more sturdy designed. Its vertical position can be adjusted so, that just a little play for the yaw bearings remains. Tailhinges are placed horizontal. An extra pipe is added into the head to facilitate the jib. Tumbler and parts of the pumprod do not have to be removed before the jib can be placed. Rotor and tail can be fitted or dismantled in any order, though it is advised to fit the rotor last, or to dismantle the rotor first. In that case there will be no rotating elements during activities. The axis of the tailvane can be adjusted until its required horizontal position is reached, by means of a turnbuckle M16.

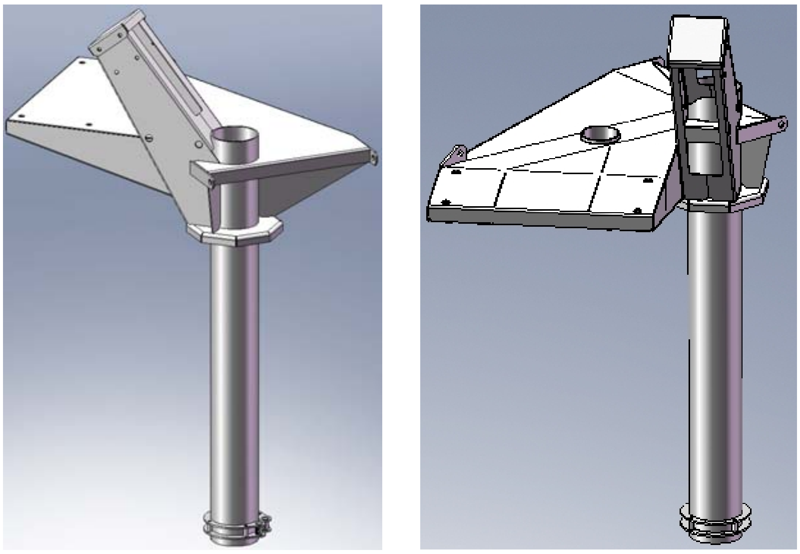


Fig. 4.2: galvanized head (sheet metal)

4.4 Jib

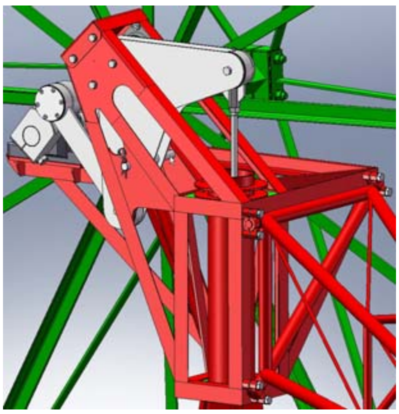
Weight: approx. 18 kg. The jib is used for fitting/ dismantling the rotor or tail onto/ from the head in case a mobile crane is not available. An already erected Diever 450 only requires dismantling when main bearings need to be replaced, or work has to be done with the vanehinge of the tail, or re-painting of the rotor is wished for. The jib is lowered into the flamepipe of the painted tower. This requires dismantling of the tumbler and top part of the pumprod. In the galvanized head an extra flamepipe is added, providing the jib can be fitted without having to dismantle these items first. So there are two different jibs, which are not interchangeable. However, their function is the same. In combination with chain hoist, hoist slings and personal safety devices (all certified!) these jobs can be carried out safely.

Fig. 4.3: Jib with two chain hoists in series for (rapidly!) hoisting the tail (photo 2005)

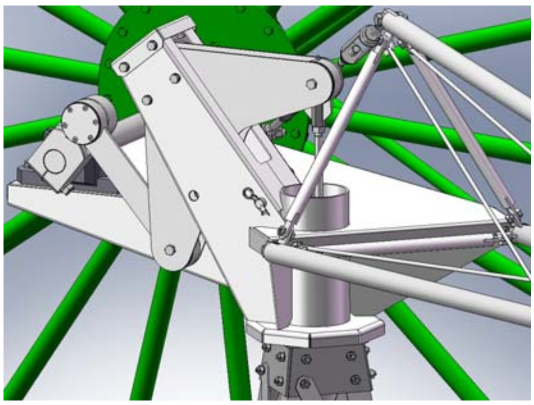
4.5 Rotation limiter

Weight: approx 0,5 kg. To lock the rotation of the rotor and transmission, and the movement of the pumprod during maintenance activities (for example: greasing of bearings, pump hauling), a rotation limiter (bar Ø 20mm) can be used. It is pushed through a hole in the head and a hole in the tumbler (or in the connecting rod, depending on type of tumbler used) of the transmission once they are aligned. One person is needed to slowly rotate the rotor by hand and another person to lock the tumbler or connecting rod. The rotation limiter has two retaining pins to prevent it from running out. The galvanized version of the head has an extra hole for storing the rotation limiter when not in use. Note: the purpose of the rotation limiter is NOT to lock the rotor rotation in harsh wind conditions, as that would invoke undesired forces in the transmission.

Fig. 4.4: rotation limiter with two retaining pins



locking, painted head,
transmission, pumprod and rotor



in storage, galvanized head,
transmission, pumprod and rotor

Fig. 4.5: locations of rotation limiter

5 Transmission

5.1 General

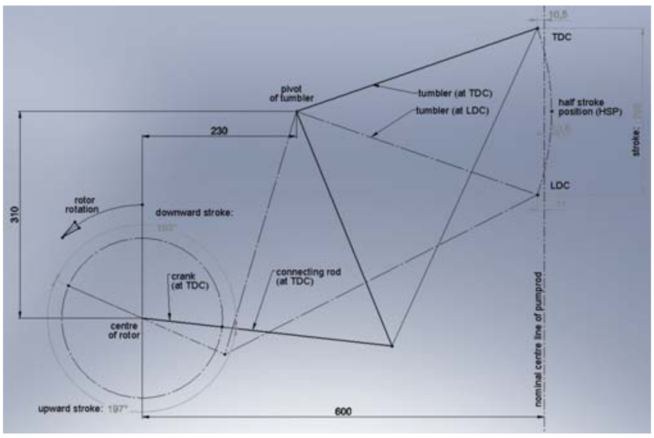
The function of the transmission is to convert the rotational movement of the rotor to a reciprocating, translating movement of the pumprod, required by the pump. In 1987, when first Diever 450 was designed, it was decided to abandon the crank, connecting rod and crosshead construction (which was used on the 12PU500) and to use a four bar mechanism with a stroke of 250mm. In 1993 the Diever 450 was furnished with an eccentric rotor and for that reason the four bar mechanism was redesigned. Stroke was set on 350mm. Because of issues with friction and maintenance, this transmission was evaluated in 2004 and recommendations were done to improve behaviour [lit. 9]. In 2016 Diever 450 on WOT testfield was equipped with transmission as proposed in 2004. A premature conclusion is that transmission seems to work fine.

photo 2004

photo 2016

Fig. 5.1: transmission

The transmission consists out of a crank, a connecting rod (conrod) and a tumbler. Weight: approx. 35 kg. (including main shaft (Ø 50h6 x 1000, material: 1.0503-C45 with bearings). The pumprod is fastened to other end of tumbler, being the fourth element of the four bars mechanism. Stroke is set on 250mm. Main shaft with its bearings is normally connected to the rotor and need not to be separated. By removing the M12 bolt (tightening torque: 90Nm) between connecting rod and tumbler, a large part of the transmission can be separated from the rotor, thus enabling fitting or dismantling of the rotor or the tumbler.



Upward stroke consumes approx. 55% of each revolution of rotor. In that way, velocities, accelerations and resulting peakforces during energy transfer from rotor to pump are minimised, see fig. 5.3.

Fig 5.2: principle of four bars mechanism applied on Diever 450

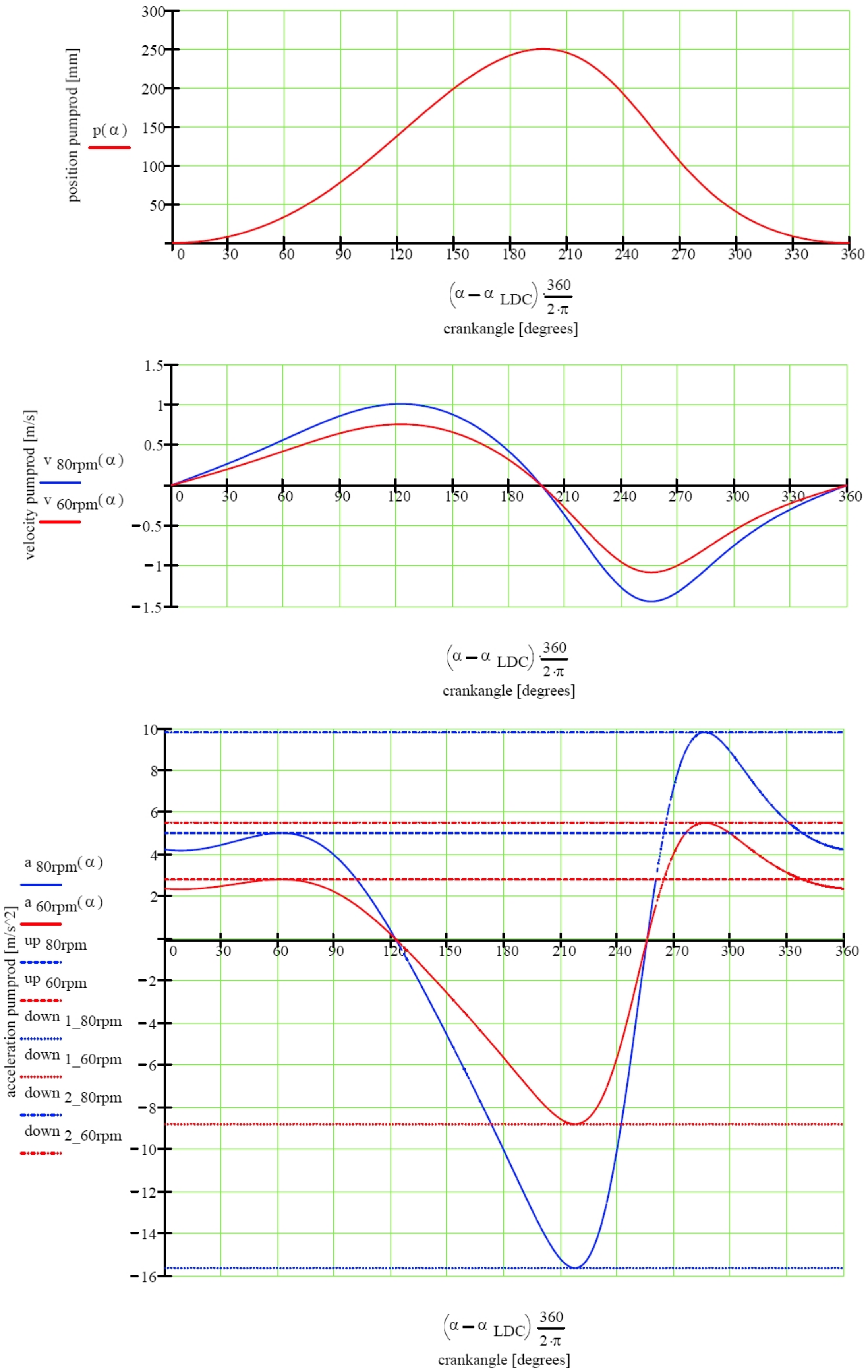


Fig 5.3: graphs of position, velocity and acceleration of pumprod at various crankangles, with regard to speed of rotor (60rpm and 80rpm)

Values of acceleration at interesting angles are given:

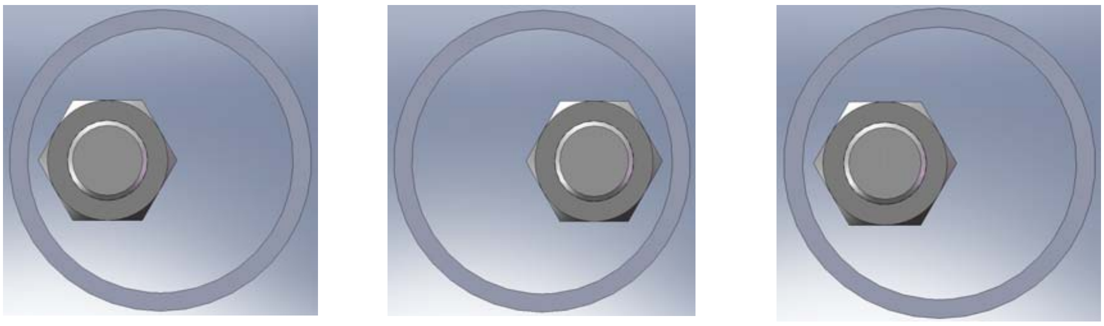
stroke	angle ($\alpha - \alpha_{LDC}$)	rotor speed: 60rpm	rotor speed: 80rpm
upward	0°	$a_{LDC\_60rpm} \approx 2{,}395 \frac{m}{s^2}$	$a_{LDC\_80rpm} \approx 4{,}258 \frac{m}{s^2}$
upward	62,0°	$a_{up\_60rpm} \approx 2{,}830 \frac{m}{s^2}$	$a_{up\_80rpm} \approx 5{,}031 \frac{m}{s^2}$
downward	217,5°	$a_{down1\_60rpm} \approx -8{,}975 \frac{m}{s^2}$	$a_{down1\_80rpm} \approx -15{,}635 \frac{m}{s^2}$
downward	286,5°	$a_{down2\_60rpm} \approx 5{,}526 \frac{m}{s^2}$	$a_{down2\_80rpm} \approx 9{,}823 \frac{m}{s^2}$

Transmission (and other parts) of Diever 450 were checked for different load cases:

- fatigue, when rotor is rotating with 60rpm. During upward stroke this results in an acceleration of pumprod, a_{up_60rpm}.
- static, when rotor is rotating with 80rpm. During upward stroke this results in an acceleration of pumprod, a_{up_80rpm}.

Curious about this approach is, though rotor is rotating at highest speed, load is being regarded as static. It is assumed: high rotorspeeds occur sporadic, so load is treated as being static. Contrary to rotor which rotates frequently at lower speeds, therefore that load is handled as fatigue.

When crank rotates, pumprod will perform a small movement from its nominal centre line. The pumprod has to enter the rising main, preferable without touching it. This goal is achieved when one uses gaspipe 2” as rising main. If one wants to avoid kling-klang noises altogether, when the Diever 450 is exposed to wind, it is advised to use gaspipe 2½" as rising main and to use very straight pumprods.



LDC HSP TDC

Fig. 5.4: pumprod (Ø16mm) with hexagon nut in rising main (2” gaspipe), top view

Note: during writing of this report, pumprod forces were checked (see chapter 11.2).
In most cases a pumprod of Ø12mm is sufficient to handle the load.
The transmission will fit the painted head and the galvanized head except for the tumbler, which needs different shaft-lengths. The two types of tumbler-shafts are therefore not interchangeable.

5.2 Static balancing of pumprod-weight

With increasing pumping depth, the weight of pumprods will increase also. To improve starting behaviour of Diever 450, pumprod-weight can be balanced by adding disks to an extended tumbler. The idea is to have the windmill set ready at TDC, waiting for (a) wind(gust). In that way, rotor can get into motion for half a revolution, without having to lift the pumprods, before the upward stroke = actual pumping stroke comes. This will improve starting at low windspeeds. Each disk balances three meter of pumprod.

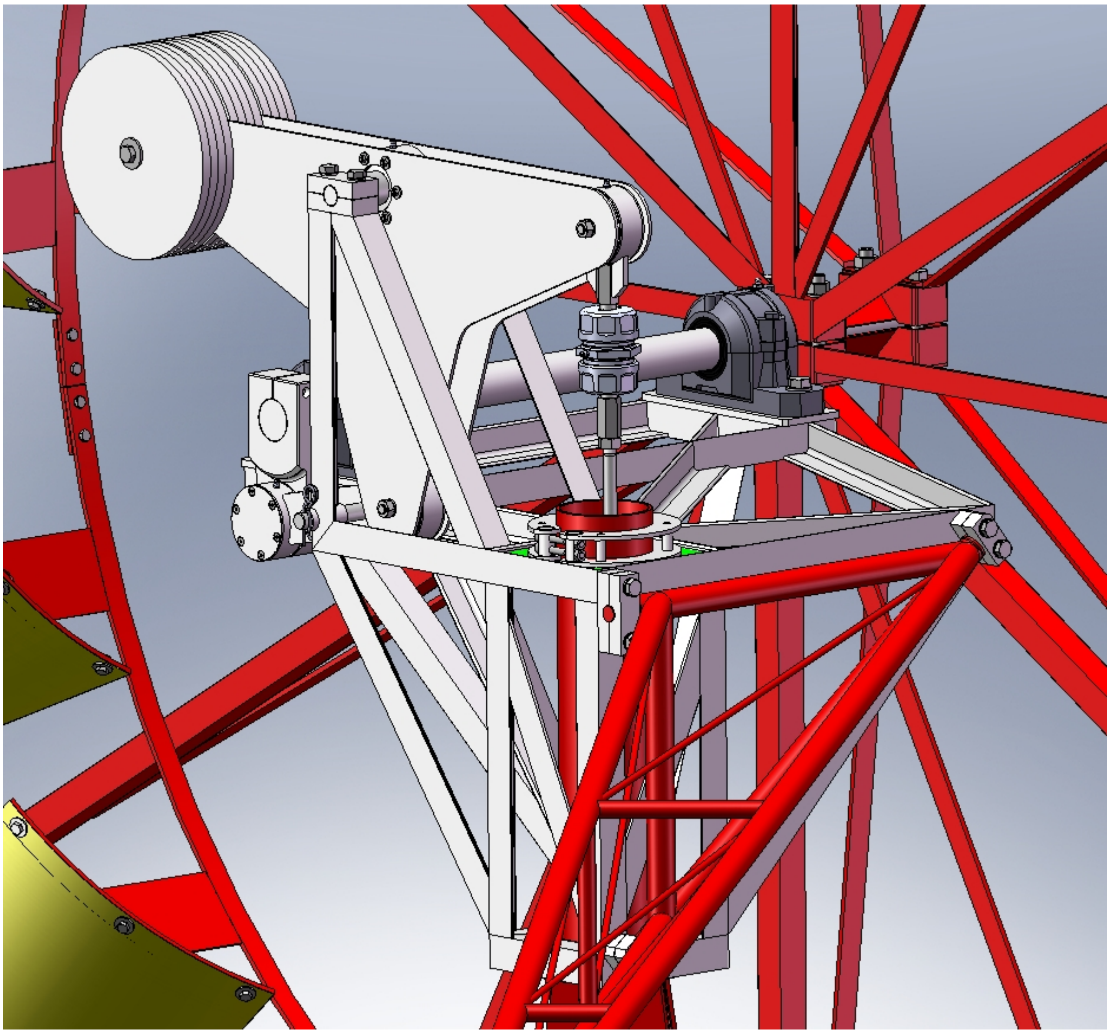
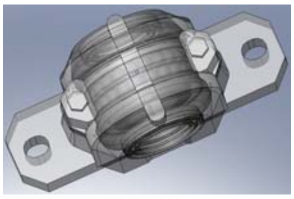


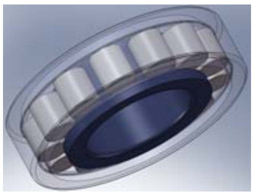
Fig. 5.5: example of balancing 36 meter of pumprods with disks

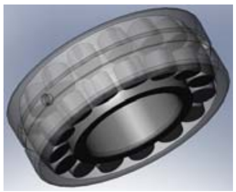
5.3 Bearings

Three types of rolling bearings are used.

- Main bearings (2x) : plummer block housing, fitted with spherical roller bearing with adapter sleeve and double lip seal. Crankside locating, rotorside non-locating.
- Tumbler bearings (2x) : cylindrical roller bearing with cage (semi locating) with labyrinth seals.
- Conrod- and pumprod bearings (3x): spherical roller bearing with labyrinth seals.


Main bearing


Tumbler bearing


Conrod- and pumprod bearing


Labyrinth seal

Fig. 5.6: bearinghousings, bearings and seals

Note: During writing of this report all bearings were recalculated. It was chosen to change the tumbler bearing (which was a radial insert ball bearing) to a cylindrical roller bearing, to increase fatigue limit load.

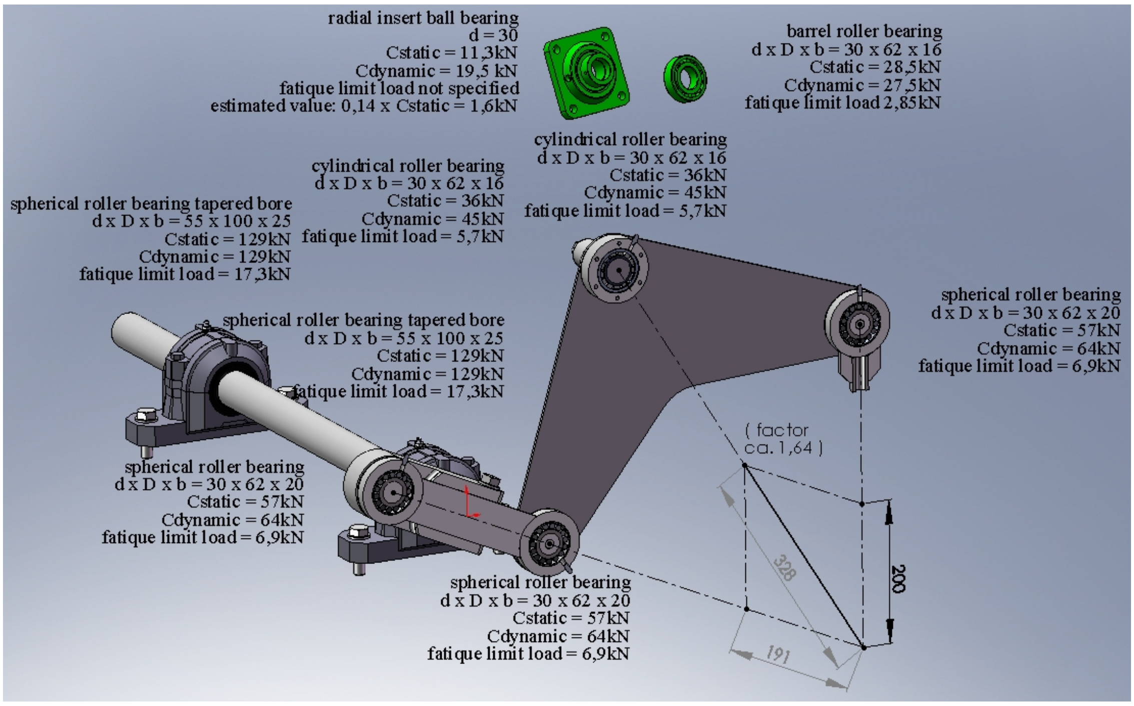


Fig. 5.7: view of bearings used. Note tumbler bearing: cylindrical roller bearing (with weaker alternatives shown in green)

All bearing housings are supplied with grease nipples. Lubrification should be done (at least once a year) with standard ball bearing grease (mineral oil as a base and lithium soap as a thickener, temperature range: -30°C till 120°C).

As some grease nipples may be difficult to reach, a grease gun with a flexible tube is advised. During re-greasing, the transmission should be locked by means of the rotation limiter to prevent jamming of (body)parts.

5.4 Crank

Weight: approx. 5,4 kg. Fabrication of the crankplate requires a milling machine including boring equipment and a (manual) rotary table. The crankpin is fabricated on a lathe. After being welded, the slot can be made with an angle grinder and the crank can be painted/galvanised. After galvanizing, check fit of bearing on the crankpin, circlip and the crank on the main shaft. Re-machine or sand down when necessary, retap threads. When the crank is being fitted to the main shaft, the M16 bolt is tightened with a torque wrench (tightening torque: 230Nm). Optional is a slightly more elegant crank, fitted with two M12 bolts (tightening torque: 93Nm).

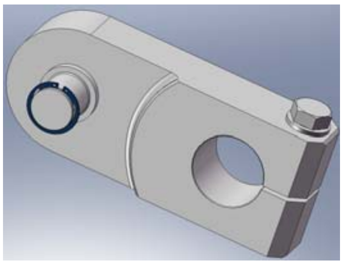


Fig. 5.8: crank fitted with one M16 bolt or two M12 bolts

5.5 Connecting rod (conrod) & pumprod bearing housing

Three identical housings which need welding. It is advised to machine holes a little undersized on a lathe before welding. After welding and galvanizing, the correct size can be achieved by boring. Tap (or retap) all threads (M5, M6, M16). Smoothen faces for covers. Six covers with δ = 3mm are needed. These can be lasercut, DXF files are available. Covers can be painted/electroplated afterwards. Internal parts, like rings and bushes, should be oiled to prevent corrosion, when stored. The connecting rod has a hole to provide entrance for the rotation limiter.

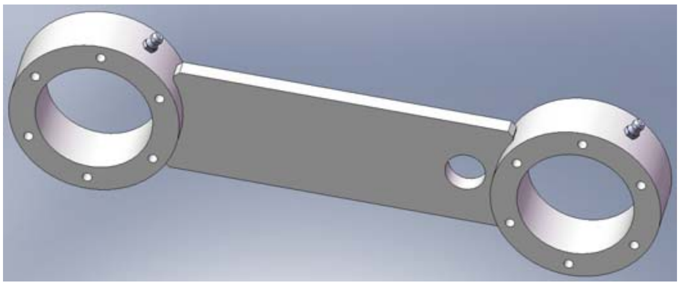


Fig. 5.9: connecting rod, weight: approx. 2,4kg

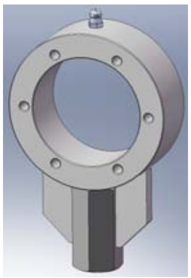


Fig. 5.10: pumprod bearing housing, weight: approx. 0,9kg

5.6 Tumbler and tumbler bearing housing

Two plates with δ = 4mm are bolted and fixated by three heavy duty spring pins to the tumbler bearing housing. The dimensions of the tumbler bearing housing are identical to the conrod- and pumprod bearing housing, except its thickness. The plates can be lasercut. DXF files are available. The length of the tumbler shaft differs for welded head and galvanized head, meaning the two shafts are not interchangeable. When assembling the conrod and the pumprod bearing housing to the tumbler, the M12 bolts must be tightened with a torque wrench (tightening torque: 90Nm).

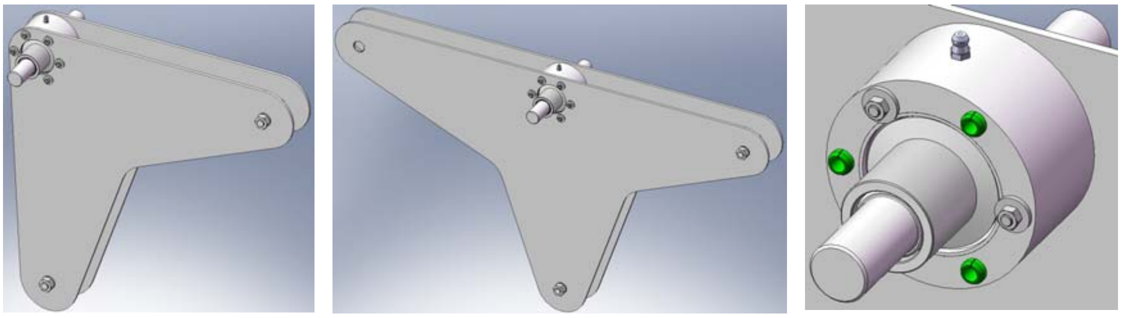
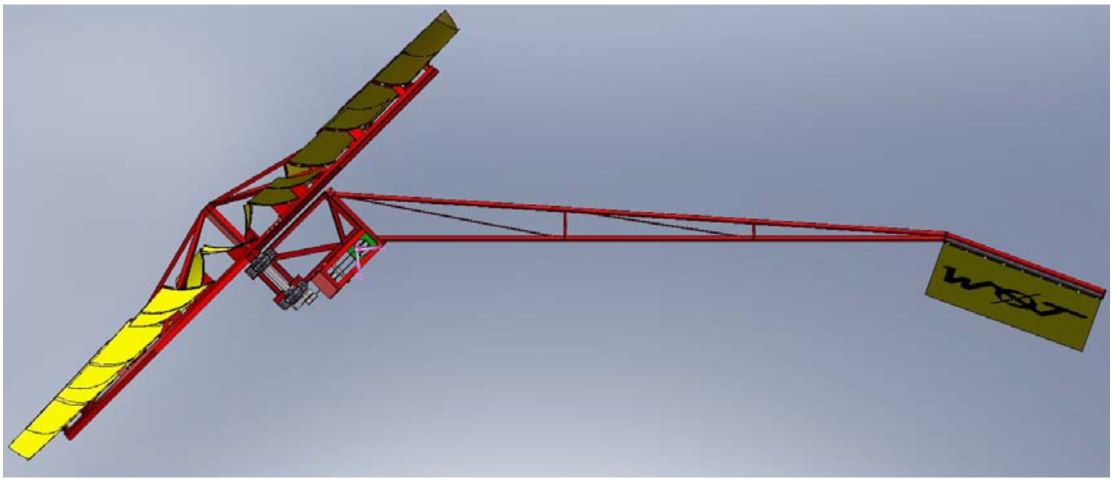


Fig. 5.11: left: tumbler, weight: approx. 9,7 kg
middle: extended tumbler for balancing, weight: approx. 13,9 kg
right: detail of connection between plates and tumbler bearing housing, heavy duty spring pins shown in green

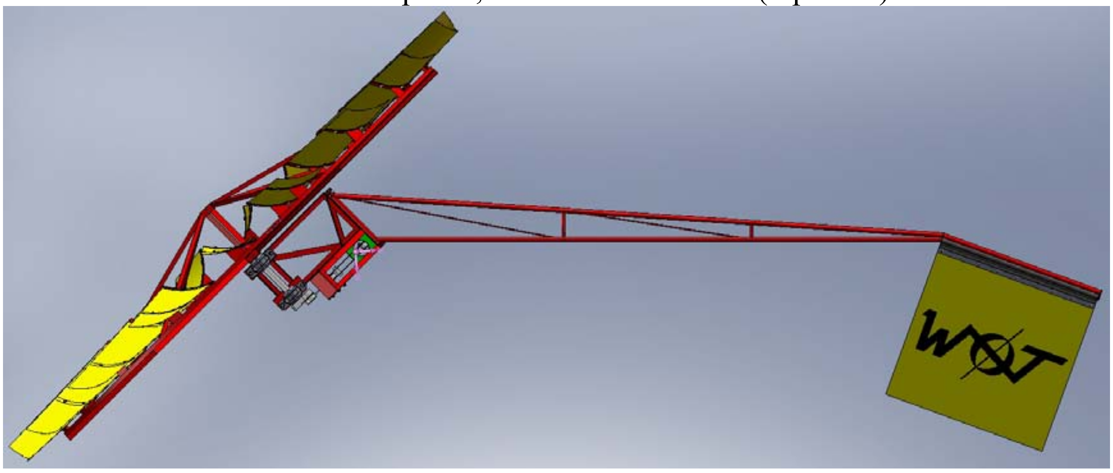
6 Tail

6.1 General

The tail (with vane) is part of the safety system which protects the rotor against too high forces and rotational speeds at high windspeeds, by turning the rotor out of the wind. In 1990 it was decided to abandon the hysteresis safety system [lit. 1] and to equip the Diever 450 with an inclined hinge main vane safety system, as described in the technical report [lit. 4]. In 1993 Diever 450 on WOT testfield got a hinged side vane as a safety system [described in lit. 2 & lit. 5 & lit. 12]. An advantage (though not much mentioned in other reports) of this system (in contrast to all other existing safety systems) is, that the centre of gravity of rotor, head, transmission and tail (when combined), will remain very close to the centre of yawing, when windspeed varies. This means yawing friction will be constant (and predictable). Risk of twisting the tower (at high windspeeds) is therefore minimised.



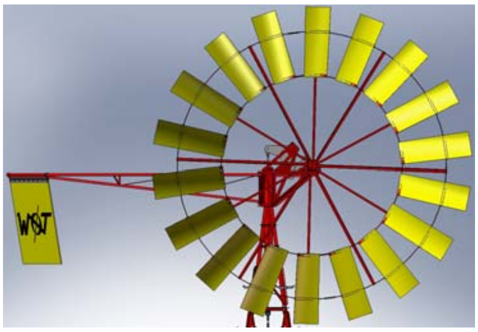
modest windspeeds, vane almost vertical (top view)



high windspeeds, vane almost horizontal (top view)

Fig. 6.1: centre of gravity of rotor, head, transmission and tail

Because the vane juts out left from the rotor, it will remain in undisturbed wind-flow, resulting in a stable behaviour of the system, even at high windspeeds.



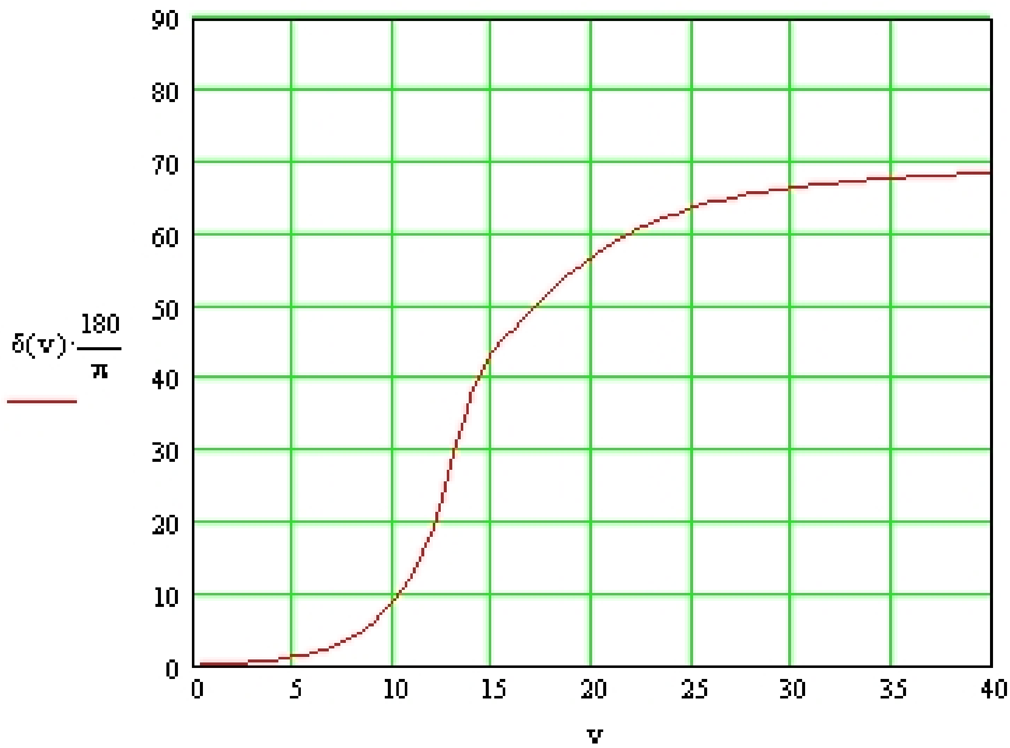
modest windspeeds:
rotor approx. 8˚ out of the wind
vane almost vertical

high windspeeds:
rotor approx. 70˚ out of the wind
vane almost horizontal

Fig. 6.2: vane in undisturbed wind-flow

Size and weight of the vane govern, for a great extent, how much rotor is turned out of the wind at a certain windspeed. In 1991 [lit. 5] calculations were made to find its best dimensions and weight. However, two years later, for prototype of Diever 450 on the testfield, it was decided to equip the tail with a lighter vane. In that way, rotor is turning out of the wind at a lower windspeed than calculated, keeping it save to observe the behaviour of the safety system. At a later stage, the vane-weight could be gradually increased, if desired. This proved not to be so, as the vane, fitted in 1993, is still in operation today. In 2005 calculations were checked [lit. 12]. The vane is made of plywood (meranti, waterproof), and needs to be painted well, before the tail is fitted to the head. Once fitted, the vane is impossible to reach. As an alternative, steel sheet (with some kind of reinforcement, to improve firmness) may be considered, as it can be galvanized. The hinge is made of sunlight-resistant rubber sheet, no maintenance. The painted tail will fit the painted head and the galvanized tail will fit the galvanized head. They are NOT interchangeable. Both tails can be revolved towards the tower, in case work needs to be done (with help of jib and chain hoist).

Vane characteristics:	Installed on prototype (since 1993)	Calculated & checked (1991 & 2005)	Steel as alternative
Dimensions [m²]	1,22 x 1,22	1,22 x 1,22	1,22 x 1,22
Thickness [mm]	12	14	1
Density [kg/m³]	640	800	7800
Weight [kg] (rounded)	11,4	16,7	11,6



predicted behaviour of safety system, when a vane is fitted with characteristics:

vane dimensions 1,22 x 1,22 [m²]

vane weight 16,7 [kg]

Fig. 6.3: angle (δ) of rotor axis in regard to winddirection at different windspeeds [lit. 12]

6.2 Painted tail

Weight (including vane): approx. 64 kg. May be difficult to transport/galvanize because its length (approx. 5,5 meter), completely welded. Right-angled triangle as base, with hinges on hypotenuse to fit the head. The tail leaves the rotor axis with 45˚. Three gaspipes (1”) converge to its tip. Braces and diagonals are welded to add rigidity. The vane is fitted at the tip, hinged. Angle between tip and tail is 20˚.

Fig. 6.4: fitting of tail: hoist sling at arrow, chain hoist on jib (shown in green), the two top-pipes of tail (axis of vane hinge) should be horizontal when the bolts (6 x M12) have been tightened (tightening torque: 90Nm)

In 1997 it was decided to improve the suspension of the vane by applying a 2½" gaspipe (instead of the former solo 1” gaspipe) at the tip. The wooden bearings of the vane were changed for a sunlight-resistant rubber sheet with δ = 1mm. (see fig. 6.5). In both tail-versions (1993 and 1997) it was possible, to manually lift the vane to horizontal and to lock it in that position. As a bonus, also the angle of rotation (swing) of the vane was limited.

Fig. 6.5: vane hinge: 2½" gaspipe, rubber sheet, vane locking device (photo 2004)

In 2016 this idea was abandoned and the provision was removed from tail of prototype on testfield. However, in January 2018, at storm conditions, it was noticed that vane did rotate 360˚ around this pipe 2½", thus unwrapping the rubber sheet, which is undesirable. Two weeks later, at another storm, the vane rotated 360˚ again. This time, the other way round, back to its original position. This problem is addressed in 2017 drawings, the tip of the tail is changed to three 1” gaspipes.

Two pipes in combination with the length of the rubber sheet will limit the swing of the vane in both directions. Prototype on testfield has not been altered yet, to be able to see if this rotating phenomenon occurs more often.

Fig. 6.6: vane hinge: 1" gaspipe, wooden bearings, vane locking device (photo 1994)

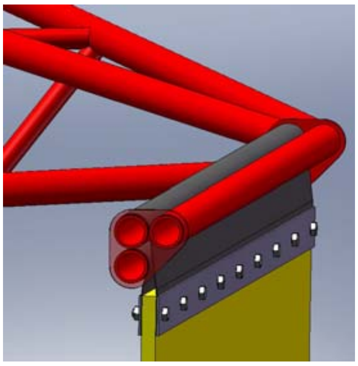


Fig. 6.7: vane hinge: sunlight-resistant rubber sheet, swing of vane is limited (a-symmetric) in both directions

6.3 Galvanized tail

Weight (including vane): approx. 80 kg. Part-lengths are set on 3 meter. Equilateral triangle as base, with hinges on horizontal to fit the head. The tail leaves the rotor axis with 45˚. Three gaspipes (1¼") converge to its tip. Braces and diagonals are bolted to add rigidity. The vane is fitted at the tip, hinged. Angle between tip and tail is 20˚.

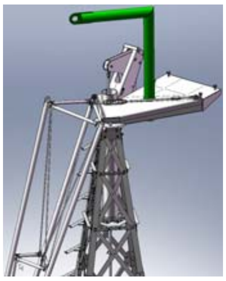


Fig. 6.8: fitting of tail: hoist sling at arrow, chain hoist on jib (shown in green), the top-pipe of tail (axis of vane hinge) must be set horizontal (with turnbuckle M16). tighten the bolts which function as hinges (2 x M16) and the counternut on turnbuckle (tightening torque: 230Nm)

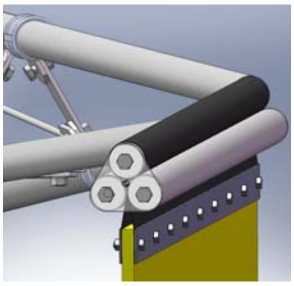


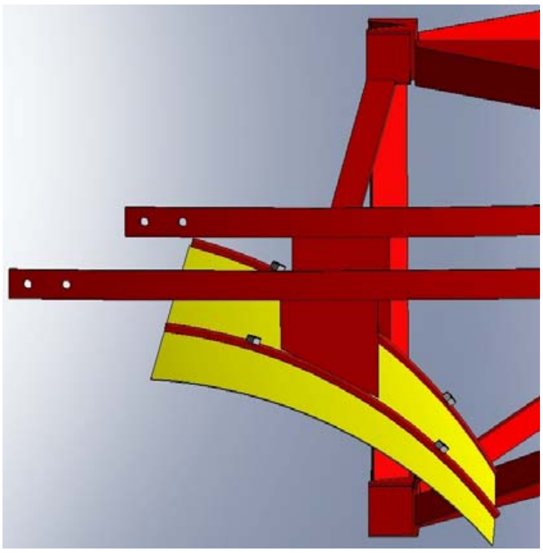
Fig. 6.9: vane hinge, sunlight-resistant rubber sheet, swing of vane is limited (symmetric) in both directions

7 Rotor

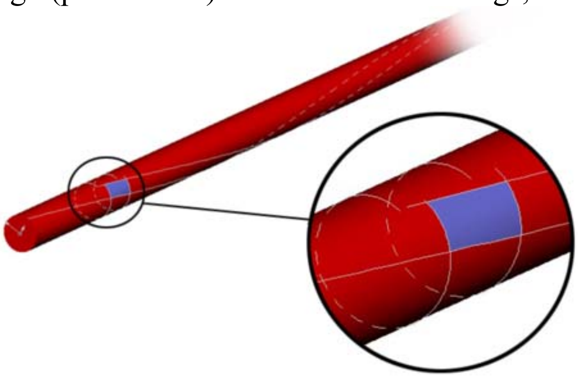
7.1 General

The rotor converts (a part of) the kinetic energy in the wind to rotating energy on its main shaft. The approach taken was to design a rotor which gives high torque at low speed (design tip-speed ratio $\lambda_d = 1$ at design windspeed $v_d = 3$ m/s). Rotor is Ø 4,5m and consists of 18 blades. Each blade (sheet metal, galvanized, $\delta = 1$mm) is cold-rolled (with radius of 500mm) has a length of 1 meter and is 400mm wide. Blades are supported at 1,25m (from rotorcenter) with angle $\beta i = 38^\circ$ and at 1,9m (from rotorcenter) with angle $\beta o = 30^\circ$. The blades are slightly twisted. According to a study carried out in 2004 [lit. 10], stress free mounting of the blade can be achieved by rolling the radius of the blade under an angle of approx. $\alpha \approx 6,2^\circ$.

twisted blade, blade-supports bolted to rings, 1987 design (photo 2004)



twisted blade, blade-supports welded to rings, 2004 design



rolling angle $\alpha \approx 6,2^\circ$ for stress free mounting of blade

Fig. 7.1: rotor

Painted rotor will fit painted head. Galvanized rotor will fit galvanized head. They are not interchangeable.

7.2 Painted rotor

When the rotor of Diever 450 was designed in 1987, experience with the rotor of 12PU500 was used. This rotor consisted out of two halves, which were clamped around the main shaft. From this clamp, six spokes fanned out to support two circular rings. Onto these rings sets of 12 blade-supports (sheet metal) were mounted. Twelve blades with a length of 2 meter and a width of 300mm were bolted to the blade-supports. Six braces were welded to give rotor strength to withstand torsion, radial- and axial forces, centrifugal moments and gyroscopic moments (which occur when rotating rotor is yawing). When transport required, rotor could be split into two (smaller) halves.

galvanized model, scale 1:2 (photo 2018)

Peru (photo 1981)

Fig. 7.2: rotor 12PU500

For Diever 450, blade-length was reduced to 1 meter and each blade (measured from tip) was placed inwards (towards rotor-center) with 250mm. Sheet-metal is normally available in size 2m x 1m. It was chosen to take 400mm as blade-width (five blades can be cut out of one sheet). The starting torque (compared with 12PU500) was maintained by adjusting the solidity of the rotor, increasing the number of blades to 18 (being dividable by 2 (halves) and 6 (spokes)). Six radial supports were added to improve ridigity. Technical Report 1990 Diever 450 states that Diever 450 has a higher starting torque than 12PU500 (starting torque: for Diever 450 ≈ 30Nm and for 12PU500 ≈ 21Nm) [lit. 4]. Weight (without main shaft, with blades): approx. 170 kg.

main shaft, clamp,
two rotor-halves assembled (photo 2005)

rotor, mounted blade-supports and blades
(photo 2005)

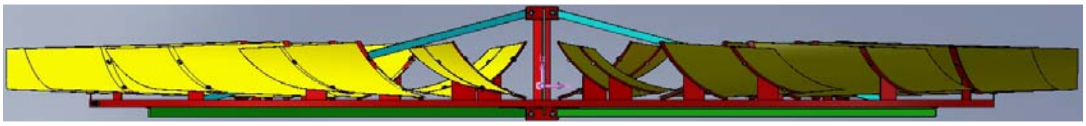
rotor and main shaft (gaspipe 2½")
(photo 2004)

rotor and main shaft (massive Ø 50h6 C45)
(photo 2017)

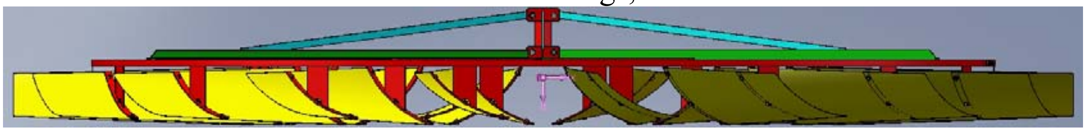
Fig. 7.3: painted rotor

In 1993 the rotor of Diever 450 on WOT testfield needed a bigger clamp to facilitate the main shaft (gaspipe 2½"). The connections between spokes (∟40.40.4) and clamp (∟50.50.5) were improved and the braces (flat 30 x 6) were turned 90˚ to improve the connections between spokes and braces. In 2016 the main shaft was changed (again) to original Ø 50mm. This time the clamp needed not to be changed. Two reducing bushes (split, to provide clamping on shaft) were fitted (without oil, grease or paint) at the location of the fasteners of the clamp. The clamp, as shown in fig. 7.5 is designed for main shaft Ø 50mm (no more need for reducing bushes). Fabrication drawings of the rotor [lit. 13] are made accordingly. When the clamp is fitted onto the main shaft the M16 bolts need to be tightened with a torque wrench (tightening torque: 230Nm). This is done on groundlevel, as the rotor is hoisted together with main shaft, plummer block housings, crank and conrod. Though not required, an extra provision at the front of the clamp, prevents the rotor sliding backwards over the main shaft.
This rotor proved very reliable. The design (in 1987) of the rotor needed no evaluation. But the observant reader may have noticed a difference of the fixation of the blade-supports to the rings. When rolling the pre-drilled rings it turned out that the rings were bent across the holes (see fig. 7.3). Although the rotorhalves will become a bit more difficult to transport, it was decided in 2004 [lit. 10] to weld the blade-supports to the rings to prevent bending, and as a bonus, saving 216 fasteners (72 bolts, 72 nuts and 72 washers).

Also, location of blades in regard of the rings was changed. This change was made in 2018, to bring the centre of gravity of rotor between the two main bearings, thus minimising radial forces in main bearings and bending moments in the main shaft.



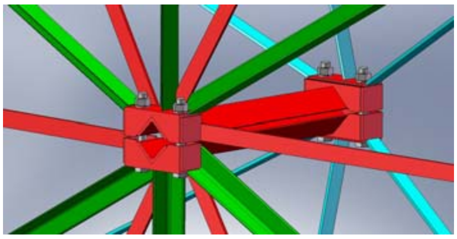
blades in front of rings, rotor 1987



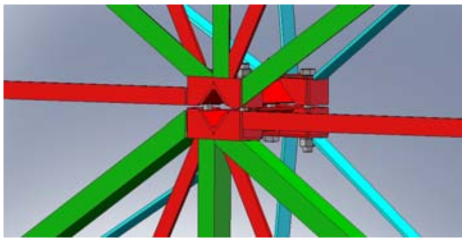
blades behind rings, rotor 2018

Fig. 7.4: centre of gravity of rotor

Spokes ∟40.40.4 are rotated 90˚.



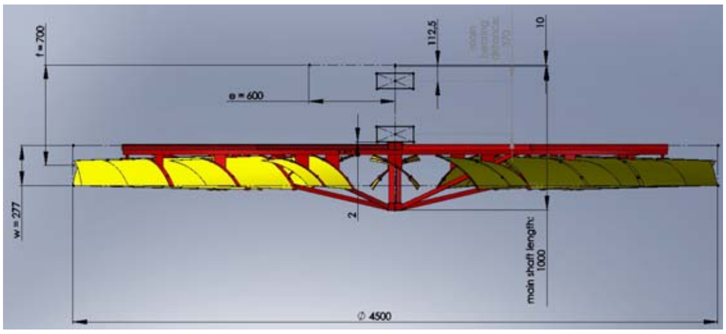
rotor 1987



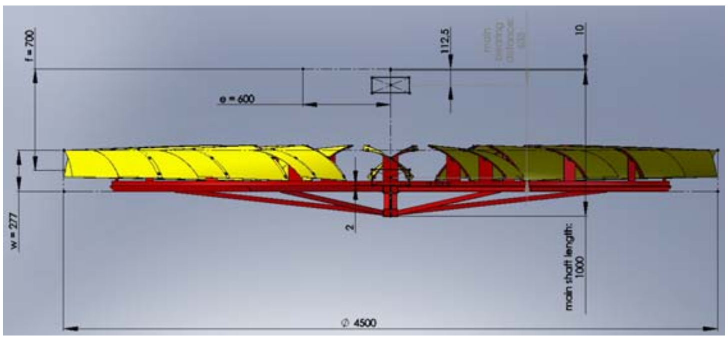
rotor 2018

Fig. 7.5: clamp & radial supports (red), spokes (green), braces (blue)

For proper functioning of safety system, distance f = 700mm needs to be maintained. For that reason, distance between main bearings is increased, resulting in a head protruding more forward.



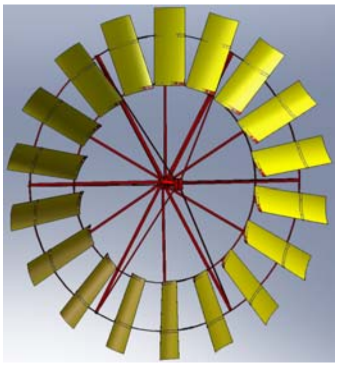
rotor 1987



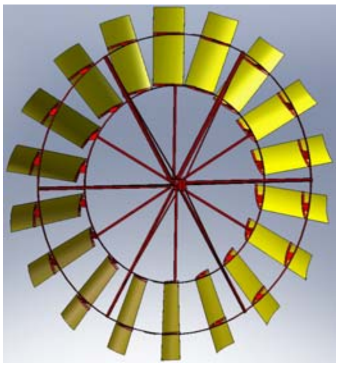
rotor 2018

Fig. 7.6: distance between main bearings

One can discuss if turbulences, caused by spokes and rings will reduce the efficiency of the rotor of 2018 when compared with the rotor of 1987. On the other hand, the rotor of 2018 has a free “behind” to let used wind go, so new wind can enter the blades. Flow-simulation or windtunneltests could provide useful information.



rotor 1987

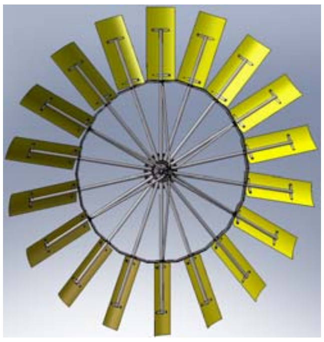


rotor 2018

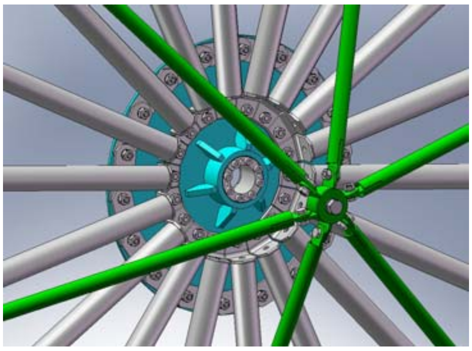
Fig. 7.7: painted rotor, front view

7.3 Galvanized rotor

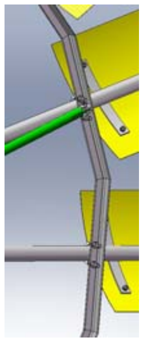
Weight: approx. 213kg. Each blade is supported by a spar (gaspipe 1¼”). These pipes are bolted to a disk. A clamping set (clampex KTR 200) is used to transfer torque to the main shaft. Spars are interconnected with ringparts using round steel U-bolts. Six braces are bolted to these ringparts converging to a nose. The nose is bolted at the end of the main shaft. Max. length of each part: 2 meter. The weight of pumprods can be balanced by adding one or more weights into the end of some spars. Realize, by doing that, the added weight will also invoke a horizontal oscillating force, when rotor is rotating. Therefore, it is preferred to balance pumprod-weight at the transmission (see chapter 5.2). This rotor has maximum efficiency because each blade is supported at the curved-in side of the blade with minimal interference (in front - or back of the blade).



front view



disk, clamping set, nose & braces



round U-bolt

spar

ringpart

counter-weight

Fig. 7.8: galvanized rotor

8 Pump

8.1 General

A single-acting reciprocating (piston)pump is used to draw water from well or borehole. During the upward stroke, water is pushed-up into the rising main by the piston with piston-valve closed. At same time, new water will flow into the pump through a foot-valve, which is opened. Around top dead centre (TDC), the foot-valve will close. During the downward stroke the piston moves (with opened piston-valve) through the water. No water is pumped during downward stroke. Around lowest dead centre (LDC) the piston-valve will close and the foot-valve will open. Both valves are operated by a combination of gravitation and flow resistance forces. The cycle is repeated as long as Diever 450 has sufficient wind.

8.2 Size

For selecting the pump size for Diever 450, at a certain mean (annual) windspeed and at a certain pumping head, the following formula can be used (according to lit. 4, updated and re-written):

$$D_c = \sqrt{5.10^5 . \frac{\pi}{g} . \frac{\rho_{air}}{\rho_{water}} . \frac{\eta_{trans}}{\eta_{pump}} . \frac{c_p}{\lambda_d} . \frac{D^3 - (D - 2.blade)^3}{stroke}} . \frac{v_{mean}}{\sqrt{H}}$$

In which:

- D_c is inner diameter of pumpcylinder [mm]
- v_{mean} is mean (annual) windspeed at location where Diever 450 is installed [m/s]
- H is pumping head of Diever 450 [m]

Around design windspeed of $v_d = 3$ m/s, values of all parameters under the (large) square root are considered being constant:

parameter	**symbol**	**value (for Diever 450)**	**[unit]**
Archimedes' constant	π	3,142	-
acceleration due to gravity	g	9,807	m/s²
density of air	ρ_{air}	1,225	kg/m³
density of water	ρ_{water}	1.10³	kg/m³
efficiency of transmission	η_{trans}	0,879	-
efficiency of pump	η_{pump}	0,9	-
power coëfficient	c_p	0,27	-
design tip speedratio	λ_d	1	-
rotor diameter	D	4,50	m
length of blade	*blade*	1,00	m
length of stroke	*stroke*	0,250	m

With these values, only valid for Diever 450, the formula can be simplified:

$D_c = \Delta_{450} \cdot \frac{v_{mean}}{\sqrt{H}}$, in which $\Delta_{450} \approx 125$ (having a awkward unit of $[m^{0,5}.s]$).

For mean (annual) windspeeds of 2,5m/s, 3m/s and 3,5m/s and various pumping heads, the ideal inner diameter of the pump can be deduced graphically:

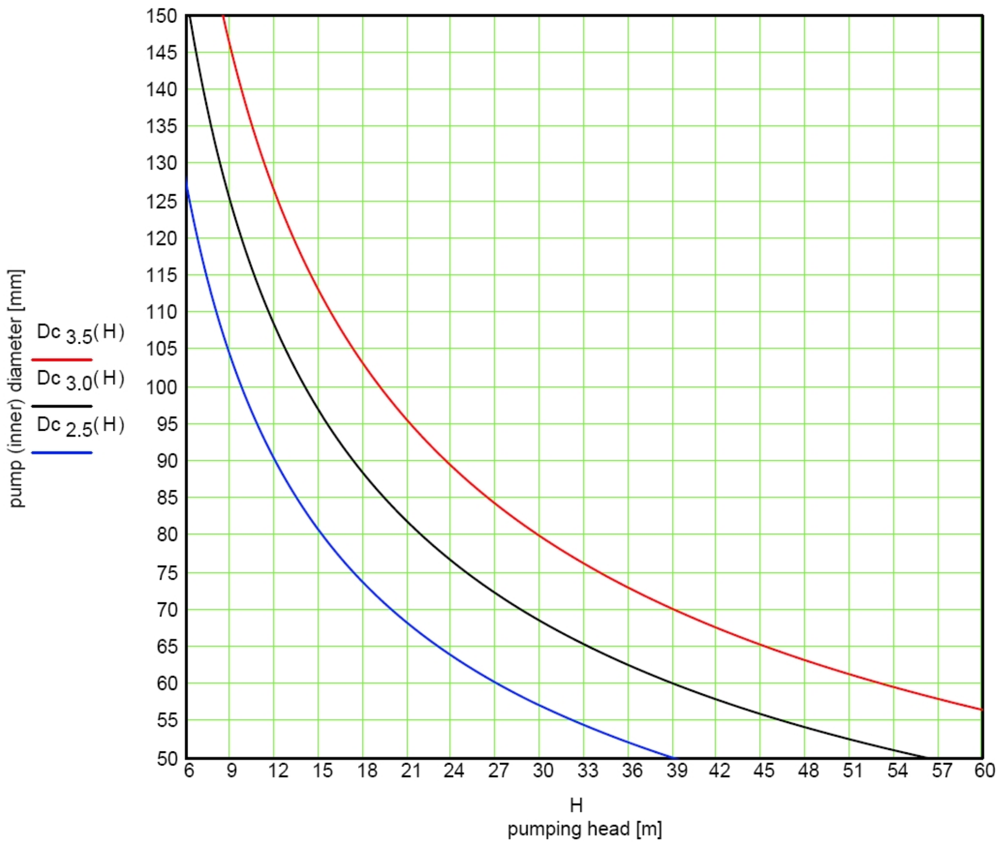


Fig. 8.1: graph for selecting pump size for Diever 450, with regard to pumping head and mean (annual) windspeed (2,5m/s, 3,0m/s and 3,5m/s)

Fig. 8.1 shows that (seemingly) minor decrease/increase in mean (annual) windspeed will have significant impact on selecting the ideal (inner) diameter of pumpcylinder for Diever 450.
Long-term windspeed-data is often collected at turbulence-free (treeless) areas, like airports, deserts, mountains, seas or oceans. Installation of Diever 450 is seldom at those places. Time, for measuring windspeeds with a anemometer, on planned site of Diever 450, often is too short to get reliable results to obtain mean (annual) windspeed.
Summarizing:

- Be cautious with external (gathered) windspeed-data
- If possible: take your own (long-term) windspeed-measurements at site, before installing Diever 450
- Check if borehole/well has sufficient capacity and take regard of water-level-drop (when pumping) to estimate pumping head
- Choose, as a start, (inner) diameter of pumpcylinder smaller than calculated, in that way you will have an output. Later on, you can decide to go for larger (in order to try gaining more output)

8.3 Output

In 2005 [lit. 12], an extensive investigation was done to predict output of Diever 450. It was considered:

- Below windspeed of 2,6m/s, there will be no output at all, as this is the starting windspeed of Diever 450. The output generated, when an already pumping Diever 450 experiences windspeeds of 2,6m/s or less, is neglected.
- Above a certain windspeed (approx. 10m/s), the windmill will furl gradually (see fig. 6.3), exposing less rotor-area to the wind, resulting in less output.
- Mean (annual) windspeed is split up (according to Weibull and Raleigh), with relevant (yearly) occurrence (see fig. 8.2). This approach will also result in reduced output.

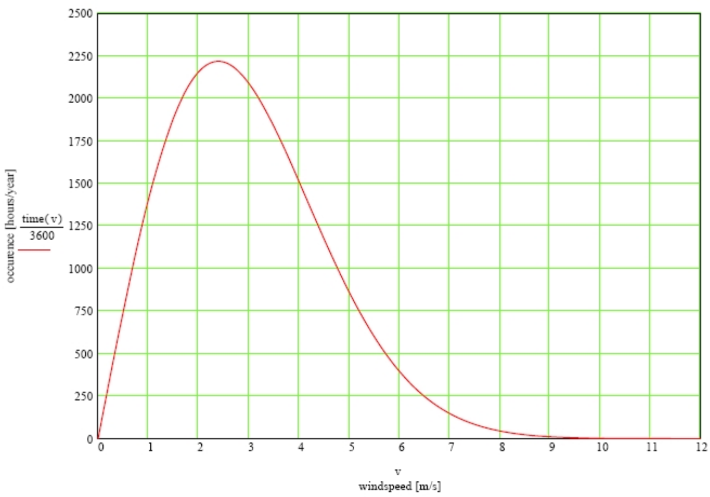


Fig. 8.2: occurrence of windspeeds assuming mean (annual) windspeed = 3m/s (Weibull and Raleigh). Example: windspeed 4m/s occurs for 1500hours/year

Calculations of 2005 were done using mean (annual) windspeed of 3m/s, with pumping head of 18m. Inner diameter of pumpcylinder was taken 83mm, resulting in an output of 19,2 m³/day. As can be deduced from fig. 8.2, for same case, the ideal inner diameter of pumpcylinder is 88,4mm. That larger diameter will result in an increased output: $\phi_{3.0}(18) = \frac{88{,}4^2}{83^2}.19{,}2 \approx 21{,}8\frac{m^3}{day}$. This value is used for generating outputs for various mean (annual) windspeeds and pumping heads. As power in wind increases with the cube of the windspeed, the output will do also. Again, a graph proves handy, to estimate output at various pumping heads and mean (annual) windspeeds:

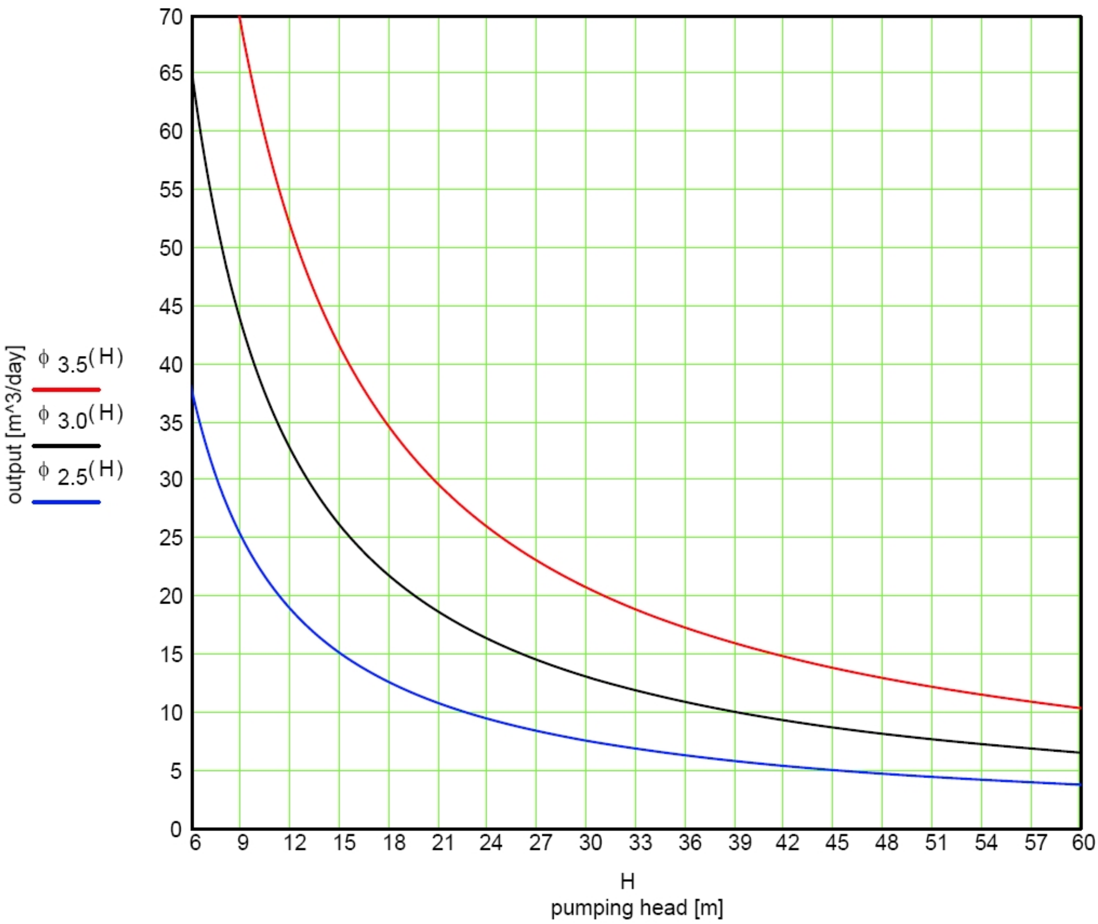


Fig. 8.3: output of Diever 450 when correct size pump has been installed, with regard to pumping head and mean (annual) windspeed (2,5m/s, 3,0m/s and 3,5m/s)

Just to illustrate uncertainty when one wants to predict the output roughly:

- If one assumes the mean (annual) windspeed is 3,0m/s, and one would say: okay, I will use that value and treat is as a constant during the year. Then the mean (annual) windspeed will stay 3,0m/s. And $v_w = 3{,}0m/s$ (all the time).
- As a result, Diever 450 will run (all year round) with a constant speed of $n = \frac{v_w.\lambda_d}{\pi.D} = \frac{3{,}0.1}{\pi.4{,}5} \approx 0{,}21s^{-1}$. In a day, that would add up to: $strokes_{day} = n.3600s.24hrs \approx 0{,}21.3600.24 \approx 18335$.

- When further assuming a pumping head of 39m, the ideal pump (inner) diameter (according to fig. 8.1) would be Ø60mm. The resulting stroke-volume is: $V_s = \frac{\pi}{4}.D_c^2.stroke = \frac{\pi}{4}.0{,}060^2.0{,}250 \approx 7{,}07.10^{-4}m^3$.
- The output would then be: $output_{day} = V_s.strokes_{day} \approx 7{,}07.10^{-4}.18335 \approx 13m^3$.
- Fig. 8.3 shows for pumping head of 39m, an output of only approx. $10m^3$.

What we found here, is a discrepancy between rough estimation of output and output-estimation according to Weibull and Raleigh. Deviation can add up to approx. 30%. Therefore, when estimating output, it is strongly advised, to use fig. 8.3 instead of above described rough-calculation.

8.4 Tests

CWD (Consultancy Services Wind Energy Developing Countries) did a lot of research to improve starting behavior of windpumps. For starting any windpump, it is good to do that with a opened piston-valve. A possibility to achieve that (as mentioned earlier in this report) is to balance the weight of the pumprods . Doing so, the rotor can get into motion for half a revolution (while water in rising-main still rests on closed foot-valve), before the actual pumping stroke comes.

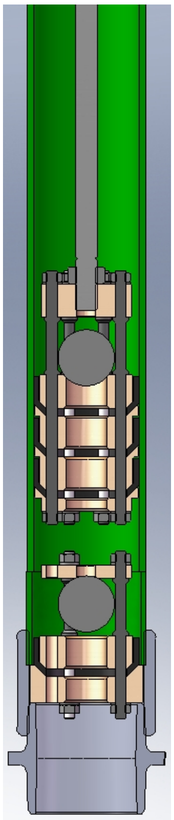
Fig. 8.4: pump-test-rig (photo 2018)

To go beyond that improvement, one could decide to close the piston-valve even later (for example 2 or 3 revolutions of the rotor, without pumping). Thus giving the rotor time to reach its ideal speed (in relation with the occurring windspeed at that moment). Combination of gravity and water-resistance forces exerted on the piston-valve governs when piston-valve will be closed. This results in a floating piston-valve; a piston-valve which will normally float in water and will be set to close by friction between waterflow and the piston-valve. Just enough to get the rotor into motion before piston-valve closes. After that, water will be lifted. WOT tried this on its pump-test-rig. The rig is placed on a tower to give easy access to pump tested. Rotational speed of the rig could be set on any speed, by means of a magnetic coupling. A fly-wheel was used to mimic rotor-inertia. Pumprod-forces were measured with strain-gauges coupled with a scope. Output of pump was measured with a (digital) water meter. Different combinations of rising mains and pumprods were installed.

A pump (2”) with a piston-valve of polypropylene (density: 946 kg/m³) was tested. The lifting height of the piston-valve was adjustable, to a be able to influence water-resistance, and so the time of closing. Unfortunately, no verifiable results could be presented to dictate/predict the ideal lifting height of the piston-valve in relation to the rotational speed. Tests about that subject were abandoned in 1998.

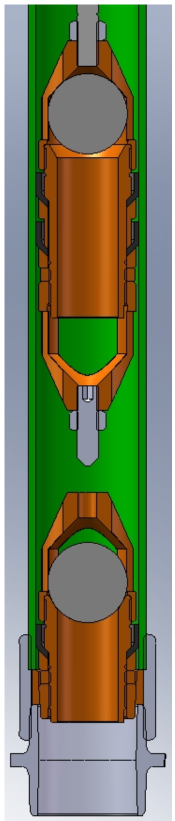
8.5 Types

WOT designed two types of pumps: a standard pump (S) and an extractable pump (E).



The standard pump is easy to machine when producing just a one-off. It uses metric stud bolts to join components of piston and foot-valve.

The extractable pump is designed for (small) series production (using brass-cast as raw material). Key feature of extractable pump is that (with smartly chosen diameter of rising mains) both piston and foot-valve can be pulled for service without having to pull (heavy) steel rising mains. Especially at deep ground water level, this feature can be a great time-saver when maintenance of pump is required.
Its foot-valve is equipped with a conical seat. The piston is furnished with a socket set screw to connect the footvalve in case work on the pump needs to be done. Piston ánd footvalve are lifted together through rising mains, using pumprods for hauling. Components of piston and foot-valve are connected with British Standard Pipe Parallel Thread (BSPP) (G), all with thread pitch 11/”. An up-to-date lathe and good craftsmanship is needed, to be able to machine an extractable pump.



standard pump (S) (photo 2017) extractable pump (E) (photo 2018)

Fig. 8.5: example of pump-types (cylinder shown in green)

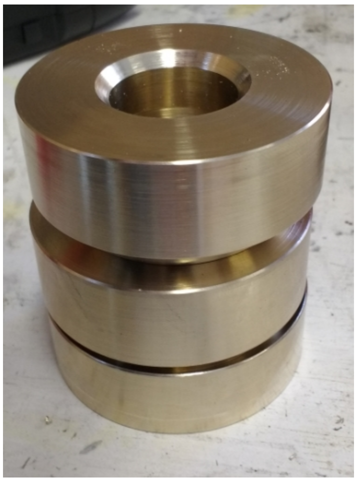
When in 2016, Diever 450 on testfield was revised, it was in need of new pump. WOT decided not buy a commercial pump, but to design and build one in-the-house. In that way gaining more knowledge/experience and once tests are satisfactory concluded, being able to offer pump-drawings to people interested.

Diever 450 on testfield has a pumping head of 12 meter (with an estimated mean (annual) windspeed of only 2,2m/s). The ideal pump-diameter is then:

$$D_c = \Delta_{450} \cdot \frac{v_{mean}}{\sqrt{H}} \approx 125.\frac{2,2}{\sqrt{12}} \approx 79,4mm$$

For the cylinder a seamless stainless steel pipe (3”) was chosen, having a inner diameter of Ø 78,5mm. Welding nipples (3” x 70) were tig-welded.
Standard pump (designed in 2016) was chosen to install under Diever 450. For the valves, stainless steel balls were used. Inner parts were machined from brass. Several leather cups for sealing the piston and one leather cup for sealing the foot-valve were implemented.
Rising main was kept 2” gaspipe, since first pump-installation in 1987.

machining of piston parts



checking fit of piston parts

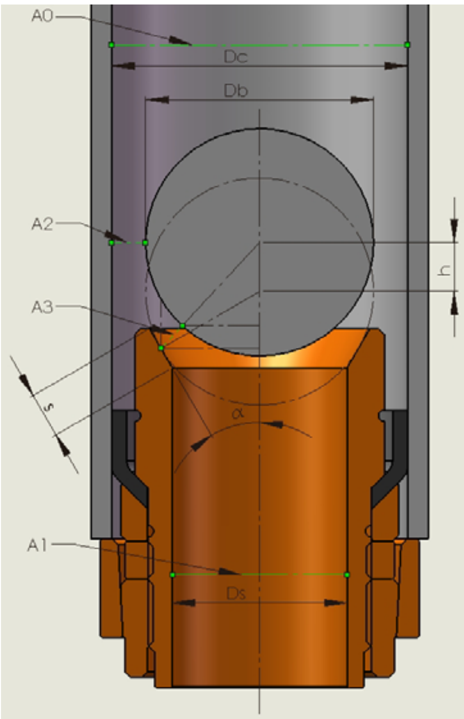
forming of leather cup

welding of cylinder

Fig. 8.6: pump fabrication (photos 2017)

8.6 Valves

Pump and especially the valves, need to be sturdy, all energy delivered by Diever 450 will be transferred in the pump. Pump is not easy to reach, once installed. One should not purchase a pump when reliability seems compromised. Valves have a double function: when closed they should seal perfectly, when opened they should restrict water-flow as little as possible. WOT decided to use balls for valves because of their nice shape. With that decision, three areas inside the pump are recognized as bottle necks, regarding water-flow. As piston and footvalve have about the same shape, only foot-valve needs to be addressed for this matter. As example, for the cylinder, the same as used under Diever 450 on testfield is taken.



D_c = inner diameter cylinder [mm]
D_b = diameter ball [mm]
D_s = diameter support m²]
h = lifting height ball [mm]
s = length seat [mm]
α = angle seat [°]

A_0 = area pump [mm²] (disk-shaped)
A_1 = area support [mm²] (disk-shaped)
A_2 = area valve [mm²] (ring-shaped)
A_3 = area lifted valve [mm²] (cone-shaped)

preset values:
$D_c = 78{,}5mm$ (pump Diever 450 at testfield)
$s = 12mm$
$\alpha = 30°$

Fig. 8.7: (extractable) footvalve with area (A_0) and constricted areas (A_1, A_2 & A_3)

To find optimized (theoretical) ball diameter (D_b), diameter of support (D_s) and lifting height of ball (h), it is considered that equal areas of A_1, A_2 and A_3 will result in equal water-velocities through those areas. For areas A_1 and A_2:

$$A_1 = A_2 = \frac{\pi}{4}.D_s^2 = \frac{\pi}{4}.\left(D_c^2 - D_b^2\right) \Rightarrow D_s^2 = D_c^2 - D_b^2 \qquad [1a]$$

Assuming ball touches seat in the middle of its length (s) when closed:

$$D_b.\cos\alpha = D_s + s.\sin\alpha \qquad [1b]$$

When combining [1a] and [1b]:

$$D_b^2 + \left(\frac{-2.s.\sin\alpha.\cos\alpha}{1+\cos^2\alpha}\right).D_b + \left(\frac{s^2.\sin^2\alpha - D_c^2}{1+\cos^2\alpha}\right) = 0 \qquad [1]$$

Filling in the preset values and using quadratic formula, the optimized ball diameter and the optimized diameter of support are found:
$D_b \approx 62{,}2mm$ and $D_s \approx 47{,}9mm$.($A_1 \approx 1800mm^2$ and $A_2 \approx 1800mm^2$).

Optimized (theoretical) lifting height (h) is found by introducing two help-variables.

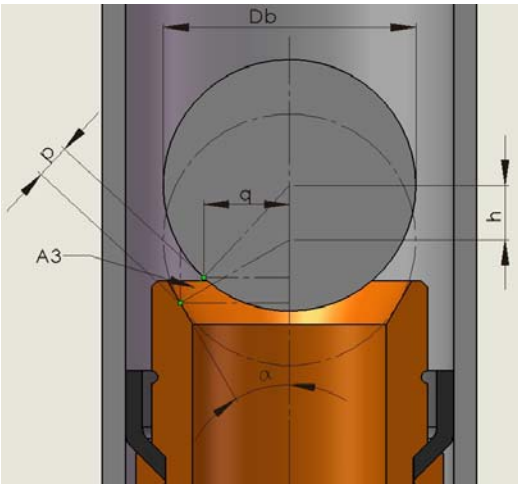


Fig. 8.8: detail footvalve with two help-variables (p & q)

Cosine rule in obtuse triangle:

$$\left(\frac{1}{2}.D_b + p\right)^2 = \left(\frac{1}{2}.D_b\right)^2 + h^2 - 2.\frac{1}{2}.D_b.h.\cos(90° + \alpha) \Rightarrow$$

$$\Rightarrow h^2 + (D_b.\sin\alpha).h + \left(-p^2 - Db.p\right) = 0 \qquad [2]$$

Use of similar (right) triangles:

$$2.q = \frac{D_b^2.\cos\alpha}{D_b + 2.p} \qquad [3a]$$

For areas A_1 and A_3 (lateral surface of a topped-off cone):

$$A_1 = A_3 = \frac{\pi}{4}.D_b\left(D_b.\cos\alpha + 2.\cos\alpha.p - 2.q\right) \qquad [3b]$$

When combining [3a] and [3b]:

$$p^2 + \left(\frac{\pi.D_b^2.\cos\alpha - 2.A_1}{\pi.D_b.\cos\alpha}\right).p + \left(\frac{-A_1}{\pi.\cos\alpha}\right) = 0 \qquad [3]$$

Filling in the earlier calculated- and preset values and using quadratic formula, the help-variables p & q are found: $p \approx 12{,}4mm$ and $q \approx 19{,}3mm$. ($A_3 \approx 1800mm^2$).
The value of p is entered into [2], again using quadratic formula and finally, the optimized lifting height is found: $h \approx 18{,}6mm$.

Concluding: water-velocity will increase, when flowing through constricted areas inside the pump. High water speeds result in hydraulic losses, high pumprod-forces and may even invoke cavitation at footvalve during upward stroke. A pump-speed-factor seems useful, $P_{sf} = \frac{A_0}{A_{1,2,3}}$, (in which $A_{1,2,3}$ is the smallest area in the pump), in order to be able to compare different designs. Note: as shown in example, even when pump is designed with smallest constrictions possible, pump-speed-factor still is:

$$P_{sf} = \frac{A_0}{A_{1,2,3}} = \frac{A_0}{A_1} = \frac{A_0}{A_2} = \frac{A_0}{A_3} \approx \frac{\frac{\pi}{4}.D_c^2}{1800} \approx \frac{\frac{\pi}{4}.78{,}5^2}{1800} \approx 2{,}7 \ .$$

Unfortunately, in practice, these optimized (theoretical) values (to keep a steady flow, resulting in smallest possible water-accelerations/ velocities, inside the pump) can only be used as a guideline, due to limited availability of ball diameters and other practical considerations. WOT came up with four pumps, which are summarized below.

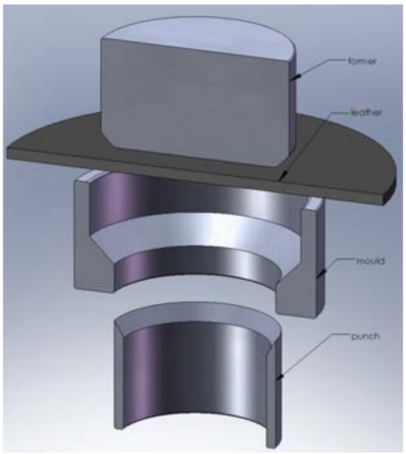
		cylinder specifications and pump-type: nominal diameter [inch] x wall thickness [mm] standard = (S), extractable = (E)			
	[unit]	2” x 5,54 (E)	2½” x 5,00 (E)	3” x 4,70 (S)	3” x 5,49 (E)
D_c	mm	49,22	66,10	78,5	77,92
A_0	mm²	1903	3432	4840	4769
theoretical values:					
s	mm	12	12	12	12
α	°	30	30	30	30
D_b	mm	40,02	52,82	62,21	61,77
D_s	mm	28,66	39,74	47,88	47,50
p	mm	6,78	10,01	12,40	12,29
q	mm	12,94	16,59	19,26	19,13
h	mm	10,43	15,13	18,61	18,45
A_1, A_2 & A_3	mm²	645	1240	1800	1772
P_{sf} (overall)	-	2,95	2,77	2,69	2,69
realized pumps:		floating piston-valve tests under pump-test-rig in 1997	designed in 2018	installed under Diever 450 at testfield in 2017	designed in 2018
s	mm	12	12	4	12
α	°	30	30	40	30
D_b	mm	31,750 (1¼”)	47,625 (1⅞”)	42,00	60,00
D_s	mm	22	36	30	46
p	mm	4,43	8,52	7,48	11,22
q	mm	10,75	15,19	11,86	18,91
h	mm	7	13	10	17
A_1	mm²	380	1018	707	1662
A_2	mm²	1111	1650	3454	1941
A_3	mm²	341	959	656	1582
P_{sf} (at A_3)	-	5,58	3,58	7,37	3,01

All four pumps, designed by WOT, show that A_3 (area of lifted valve) is less than other constricted areas (A_1 and A_2) inside the pump. The area of lifted valve can be influenced by changing lifting height of valve (h). The idea is to choose lifting height so, that most water-resistance will be generated at location of the valve, which will help to close the valve as fast as possible.

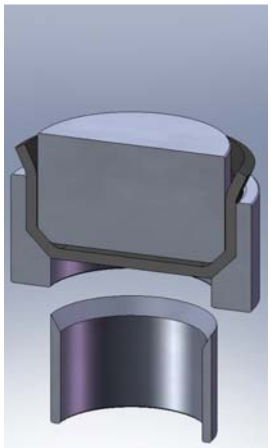
Apart from minor issues with leather cups, pump (3” x 4,70 (S)), installed under Diever 450 at testfield in 2017, works fine, although pump-speed-factor seems relatively high.

8.7 Seals

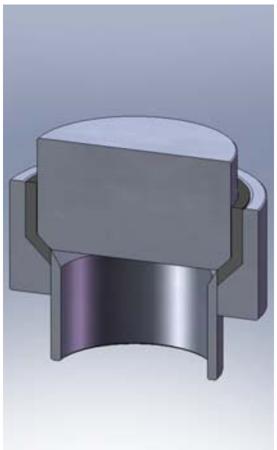
For sealing the gap between piston/foot-valve and cylinder, leather cups are used. When fabricated and mounted correct, these leather cups can have a long life-span, probably because they are always lubricated with water.
It is obvious, when sand-particles are sucked up by the pump, the life of cups will shorten and damage to valve-seats may occur. For that reason, the pump should be installed way above bottom of well/borehole.
Some clearance between cups and supports (see fig. 8.7) is necessary. Cups have the tendency to swell when submerged in water, despite being impregnated with candlewax. During pumping, waterpressure will push the cup against the inner-side of cylinder, sealing the gap.



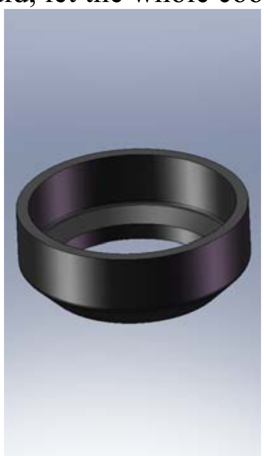
soak leather, δ = 4mm, in candlewax of approx. 90°C until no more airbubbles come out



press leather with former into mould, let the whole cool off



punch hole into cup with punch, trim excess of cup with sharp knife



cup, ready to be mounted on piston or foot-valve

Fig. 8.9: example of cup-fabrication

9 Foundation

9.1 General

In 1990 [lit. 4] forces in towerlegs were calculated for Diever 450 with inclined hinge main vane with sidevane, having a hub height of approx. 11 meter. Highest values were found when:

- severe wind conditions (at survival windspeed = 40m/s) occur and,
- rotor is 90˚ out of the wind (not rotating) and,
- main vane and side vane are parallel to winddirection and,
- head is not yawing and,
- tower is struck diagonally by the wind

Despite weight of Diever 450 (tower, head, transmission, rotor, main vane and side vane) and weight of rising main, windmill wants to tip over at severe wind conditions. In other words, tower is trying to lift one foundation block and trying to push other three foundation blocks further into the ground. Tensile force in towerleg was calculated at 13,3kN for Diever 450, when equipped with inclined main vain with side vane. As winddirection at storm conditions is unknown, all foundation blocks must be equally heavy to withstand tensile forces in towerlegs. When assuming there is no friction between foundation blocks and surrounding soil, and assuming concrete density is approx. 2200kg/m³, minimum volume of each foundation block must be approx. 0,62m³. As example: when a cube is taken as foundation block, each edge should be minimal 0,85m (≈ 90cm, when rounded).

In 1993 Diever 450 on testfield got equipped with a hinged side vane with an eccentric rotor. It behaves very different than inclined main vane safety system, although their function is the same.

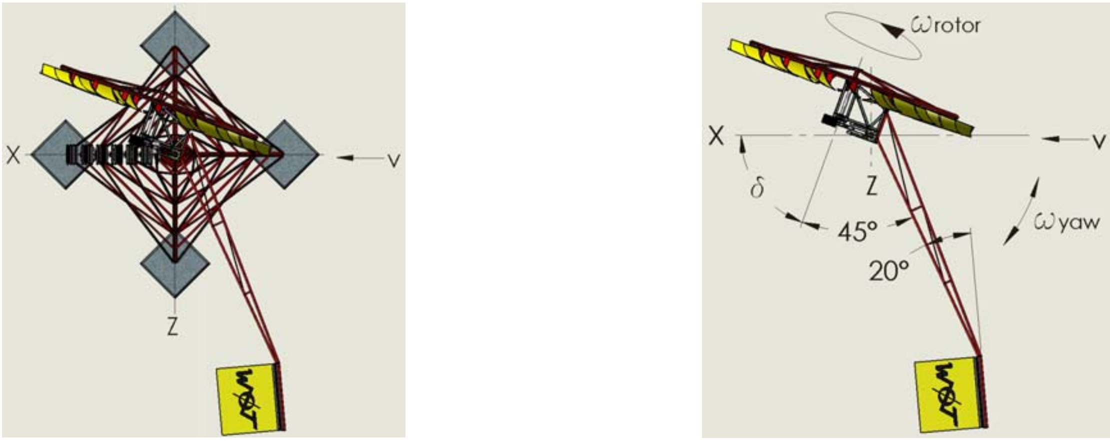
- vane horizontal
- rotor rotating with approx. 8,4 rad/s
- head yawing counter-clockwise (seen from top) with approx. 1,5 rad/s
- pump NOT disconnected

Fig. 9.1: behaviour Diever 450 at storm conditions: windforce 9, windgusts of 33m/s at testfield (snapshot of film made 18 January 2018)

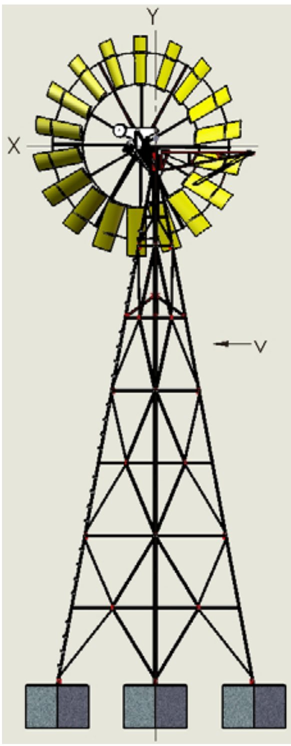
Earlier calculations of forces in towerlegs might therefore not be accurate enough. It is good to re-calculate forces in towerlegs for this safety system, to estimate minimum volume of foundation blocks.

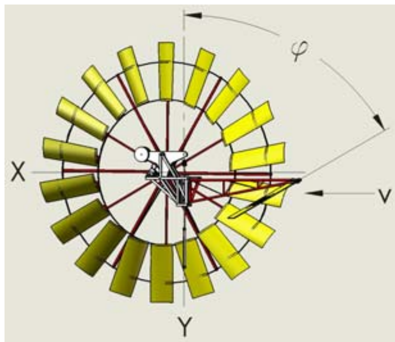
In all likelihood, highest values will be found when:

- severe wind conditions occur ($v \approx 40m/s$ (survival windspeed)) and,
- rotor is $\delta \approx 70°$ out of wind (lit. 12) and,
- rotor is rotating with $\omega_{rotor} \approx 8{,}4rad/s$ (lit. 4) and,
- piston of pump and most parts of pumprods are, most likely, disconnected by load limiter and,
- vane is parallel to winddirection ($\varphi \approx 90°$) and,
- head is yawing with $\omega_{yaw} \approx 0{,}63rad/s$ (lit. 4), due to changes in wind-direction and,
- tower is struck diagonally by wind (lit. 4).

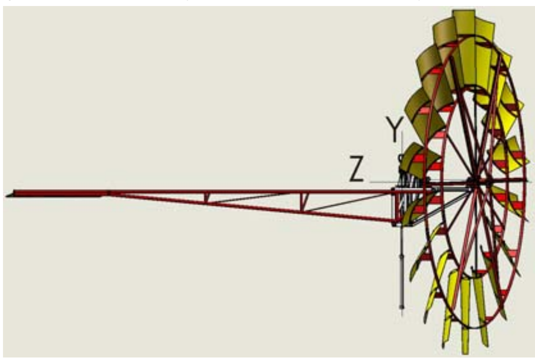


coordinate system with parameters





φ not correctly scaled, for clarity reasons



tower, struck diagonally by wind

situation "seen" by wind ($\delta \approx 70°, \varphi \approx 90°$)

Fig. 9.2: sketches of Diever 450 at severe wind conditions

9.2 Calculations

Tower experiences forces and moments during severe wind conditions:

- gyroscopic moment
- weight of rotor, head, transmission, tail and vane
- windforce on rotor, head, transmission, tail and vane
- friction in yaw-bearings, when head is yawing
- weight of tower, rising main (filled with water), most pumprods, pump, etc.
- windforce on tower and parts of rising main

These forces and moments need to be addressed, to be able to find reaction forces of foundation blocks.

9.2.1 Gyroscopic moment of rotor, main shaft, crank and part of conrod

Rotating rotor acts like a compass: resisting yawing. When rotor is rotating around rotoraxis ánd yawing around towerpipe, a gyroscopic moment is generated: $M_{gyr} = I_p.\omega_{rotor}.\omega_{yaw}$. In which I_p is moment of inertia of rotor. I_p is found when adding (vectorial) moments of inertia from two planes, which are orthogonal, and both going through centre of mass: $I_p = \sqrt{L_{xx}^2 + L_{yy}^2}$.

Main shaft, crank and part of conrod are rotating as well. Though probably not significantly influencing gyroscopic moment, these elements are added to the model. L_{xx} , L_{yy} and mass are found graphically:

Fig. 9.3: Inertia and mass of rotating-and yawing elements

Radius of inertia (i) is defined: square root of moment of inertia divided by its mass:

$$i = \sqrt{\frac{I_p}{mass}} = \sqrt{\frac{\sqrt{L_{xx}^2 + L_{yy}^2}}{mass}} \approx \sqrt{\frac{\sqrt{199^2 + 201^2}}{193}} \approx 1{,}21m$$ (just for info: not further used).

Gyroscopic moment of rotating- and yawing elements:

$$M_{gyr} = \sqrt{L_{xx}^2 + L_{yy}^2}.\omega_{rotor}.\omega_{yaw} \approx \sqrt{199^2 + 201^2}.8{,}4.0{,}63 \approx 1499\,Nm$$

When factorizing gyroscopic moments of elements, according to actual position ($\delta \approx 70°$) of rotor in relation to coordinate system as described in fig. 9.2:

$$Mx_{gyr} = M_{gyr}.\sin(\delta) \approx 1499.\sin(70°) \approx 1409\,Nm$$

$$Mz_{gyr} \approx M_{gyr}.\cos(\delta) \approx 1499.\cos(70°) \approx 513\,Nm$$

Depending on direction of yawing, both moments can switch from positive to negative.

9.2.2 Weight of rotor, head, transmission, tail and vane

Weight of rotor, head, transmission, tail and vane are combined. Its line of action is parallel with Y-axis and is named: G_{rhttv} . Transmission is supplied with an extended tumbler for balancing 36 meter of pumprods. Piston of pump and most pumprods have been disconnected by load limiter (subject will be discussed later in this report).

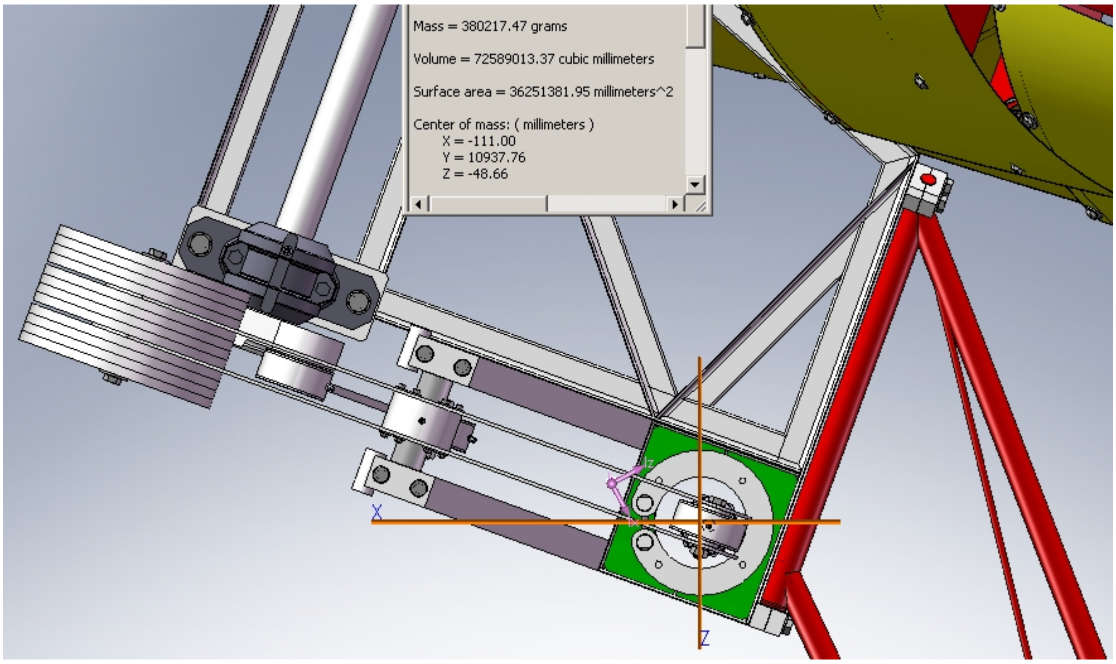


Fig. 9.4: mass of rotor, head, transmission, tail and vane

$G_{rhttv} = Mass.g \approx 380.9{,}807 \approx 2723N$

Because line of action is not central to coordinate system, two moments are generated:

$Mx_{rhttv} = G_{rhttv}.Z \approx 2723.0{,}04866 \approx 181Nm$

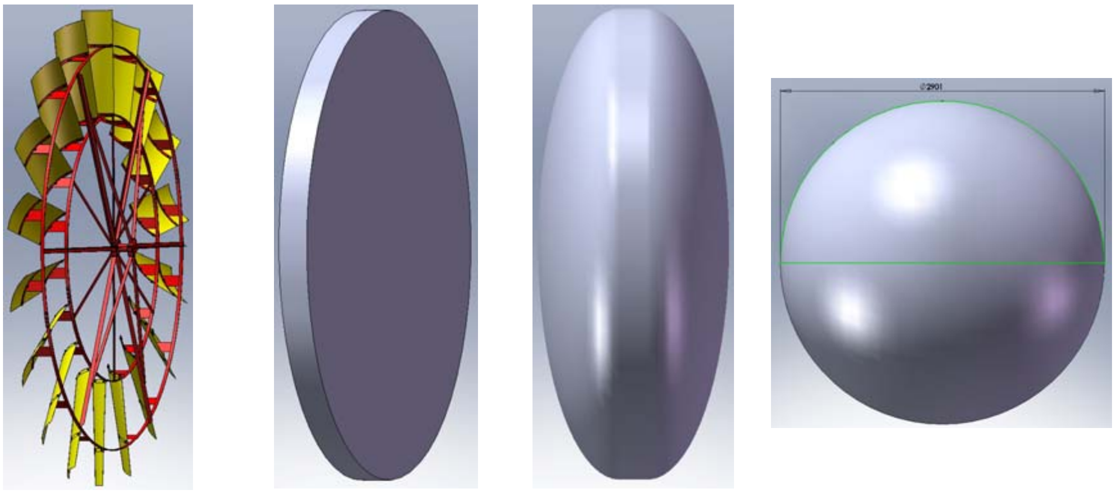
$Mz_{rhttv} = G_{rhttv}.X \approx 2723.0{,}11100 \approx 413Nm$

9.2.3 Windforce on rotor, head, transmission, tail and vane

Fig. 9.2 shows head and transmission are in wake of rotor. Vane is parallel to winddirection ($\varphi \approx 90°$). Therefore, windforces on head, transmission and vane are neglected.

Rotor exposes its front and its side to the wind. Angle (δ) influences wind-struck areas. Assumed is:

- rotor is simplified to massive sphere, having same projected area as rotor ($Ar_{projected} \approx 6{,}61 m^2$ when $\delta \approx 70°$) and,
- projected area of rotor is generating drag-force and,
- rotor is NOT generating thrust-force, as pump has been disconnected.
- Drag coefficient (c_d) of sphere is approx. 0,47.

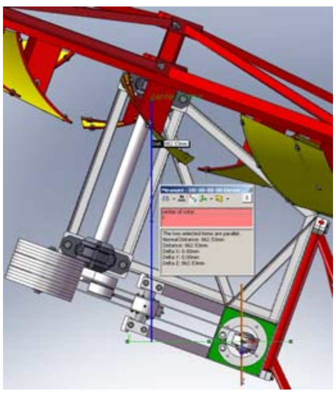


rotor (2018) with $Ar_{projected}$	disk with $Ar_{projected}$	two ellipses with $Ar_{projected}$	sphere with $Ar_{projected}$

Fig. 9.5: rotor “seen” by wind ($v \approx 40 m/s$, $\delta \approx 70°$) and simplified

Windforce on rotor (F_{rotor}) is:

$$F_{rotor} = \frac{1}{2}.\rho_{air}.v^2.c_d.Ar_{projected} \approx \frac{1}{2}.1{,}225.40^2.0{,}47.6{,}61 \approx 3045 N$$



Line of action is parallel to X-axis. Assumed is, its point of application is on axis of rotor and in the middle of its width (w). Windforce on rotor (F_{rotor}) generates a moment around Y-axis:

$$My_{rotor} = F_{rotor}.\Delta Z \approx 3045.0{,}863 \approx 2628 Nm$$

Fig. 9.6: point of application of windforce on rotor

Line of action of windforce on tail (F_{tail}) is parallel to X-axis and point of application is in middle of projected tail. Note: tail is NOT perpendicular to winddirection.

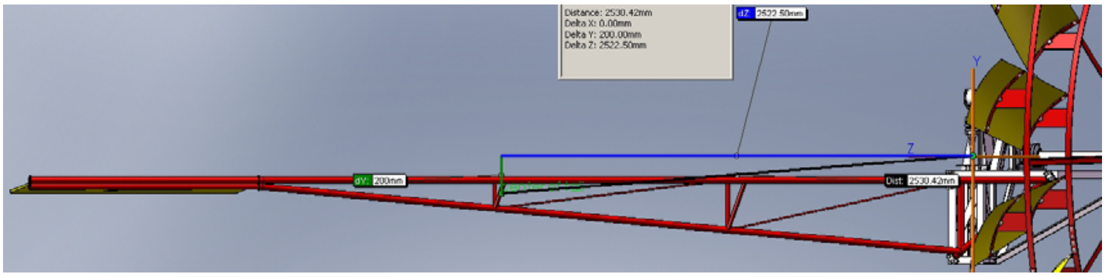


Fig. 9.7: point of application of windforce on tail

Windforce on tail (F_{tail}) is trying to push rotor back into the wind, but is not succeeding. In other words: resulting moment around Y-axis is zero (friction in yaw-bearings is neglected at this stage).

In this way F_{tail} can be found:

$$My_{rotor} = My_{tail} = F_{tail}.\Delta Z \Rightarrow 2628 \approx F_{tail}.2{,}523 \Rightarrow$$

Windforce on tail: $F_{tail} \approx \dfrac{2628}{2{,}523} \approx 1042N$

Windforce on tail (F_{tail}) also generates a moment around Z-axis:

$$Mz_{tail} = F_{tail}.\Delta Y \approx 1042.0{,}200 \approx 208Nm$$

9.2.4 Friction in yaw-bearings

Two yaw bearings, made of HMPE, keep head fixated around (steel) towerpipe. Static frictional coefficient between those two materials is set on: $\mu \approx 0{,}2$. HMPE has a compressive strength of approx. $\overline{\sigma d}_{HMPE} \approx$ 32N/mm². Lower bearing (L) transmits radial- and axial loads, upper bearing (U) transmits only radial load. Distance: $g = 99mm$ and distance: $h = 458mm$.

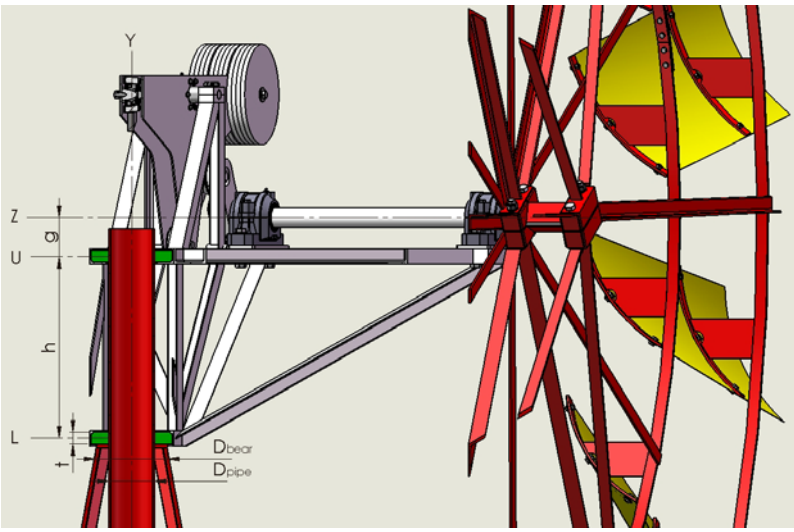


Fig. 9.8: two yaw bearings, keeping head fixated around towerpipe

Forces and moments, already calculated, must be distributed over two yawing bearings, each with thickness $t \approx 30mm$. Towerpipe has diameter $D_{pipe} \approx 114{,}3mm$. Bearing plate has diameter of $D_{bear} \approx 180mm$.

For X-axis:

$$Fx_U = \frac{Mx_{gyr} + Mx_{rhttv}}{h} + \frac{g+h}{h}.(F_{rotor} + F_{tail}) \Rightarrow$$

$$Fx_U \approx \frac{1409+181}{0{,}458} + \frac{0{,}099+0{,}458}{0{,}458}.(3045+1042) \approx 8442N$$

$$Fx_L = \frac{Mx_{gyr} + Mx_{rhttv}}{h} + \frac{g}{h}.(F_{rotor} + F_{tail}) \Rightarrow$$

$$Fx_L \approx \frac{1409+181}{0{,}458} + \frac{0{,}099}{0{,}458}.(3045+1042) \approx 4355N$$

For Z-axis:

$$Fz_U = Fz_L = \frac{Mz_{gyr} + Mz_{rhttv} - Mz_{tail}}{h} \approx \frac{513+413-208}{0{,}458} \approx 1568N$$

For Y-axis:

$Fy_U = 0$, and $Fy_L = G_{rhttv} \approx 2723N$

Upper yaw bearing generates friction (W_U) of:

$$W_U = \mu.\sqrt{Fx_U^2 + Fz_U^2} \approx 0{,}2.\sqrt{8442^2 + 1568^2} \approx 1717N.$$

Resulting in a frictional moment in upper yaw bearing (M_U) of:

$$M_U = W_U.\frac{D_{pipe}}{2} \approx 1717.\frac{0{,}1143}{2} \approx 98Nm$$

Lower yaw bearing generates radial friction (Wr_L) of:

$$Wr_L = \mu.\sqrt{Fx_L^2 + Fz_L^2} \approx 0{,}2.\sqrt{4355^2 + 1568^2} \approx 925N$$

Resulting in a radial frictional moment in lower yaw bearing (Mr_L) of:

$$Mr_L = Wr_L.\frac{D_{pipe}}{2} \approx 925.\frac{0{,}1143}{2} \approx 53Nm$$

Lower yaw bearing also generates axial friction (Wa_L) of:

$$Wa_L = \mu.Fy_L \approx 0{,}2.2723 \approx 545N$$

Assuming axial friction lies on average diameter of towerpipe and diameter of bearing plate, axial frictional moment in lower yaw bearing (Ma_L) is:

$$Ma_L = Wa_L.\frac{D_{pipe} + D_{bear}}{4} \approx 545.\frac{0{,}1143 + 0{,}180}{4} \approx 40Nm$$

Tower experiences a moment (due to friction) around Y-axis ($M_{friction}$) of:

$$M_{friction} = M_U + Mr_L + Ma_L \approx 98 + 53 + 40 \approx 191Nm$$

9.2.5 Weight of tower, rising main (filled with water), most pumprods, pump, etc.

Weight of tower, rising main (filled with water), most pumprods, pump and other (small) items are combined.

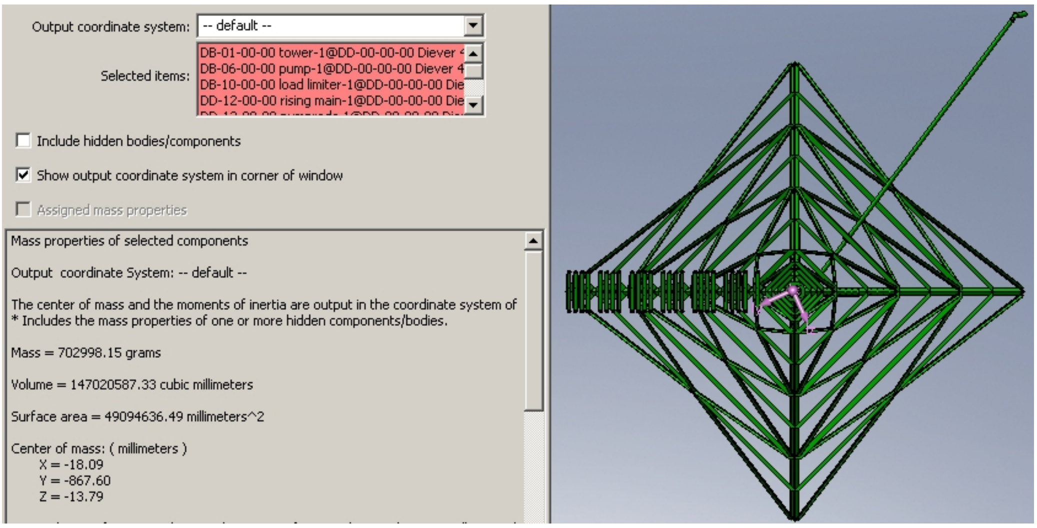


Fig. 9.9: weight of tower, rising main (filled with water), most pumprods, pump and other (small) items

Note: centre of mass ($m_{tower\&etc}$) has shifted below base of tower ($Y \approx -868$ mm), because pump is mounted 36 meter below actual XZ-plane (as shown in fig. 9.2).

Water in rising main: fig. 5.4 shows pumprod, $d_{pr} \approx 16mm$, inside rising main (2" gaspipe, having a inner diameter of approx. $Di_{rm} \approx 53mm$). Assumed is, delivery pipe of rising main is 5 meter below XZ-plane. Mass of water in rising main (m_{water}) is:

$$m_{water} = \rho_{water}.\frac{\pi}{4}\left(Di_{rm}^2 - d_{pr}^2\right)height \approx 1000.\frac{\pi}{4}.\left(0{,}053^2 - 0{,}016^2\right)(36-5) \approx 62kg$$

Mass of each item is given:

item	mass [kg]	item	mass [kg]
tower with steps	393,10	(part of) load limiter	2,40
rising main (2")	178,07	vertical limiter	1,34
most pumprods	51,25	pumprod lock	0,55
pump	14,14	water in rising main	62,16

total mass of tower & etcetera ($m_{tower\&etc}$) is approx. 703kg.

Weight of tower & etcetera ($G_{tower\&etc}$) is:

$$G_{tower\&etc} = m_{tower\&etc}.g \approx 703.9{,}807 \approx 6894N$$

Because line of action is not central to coordinate system, two moments are generated:

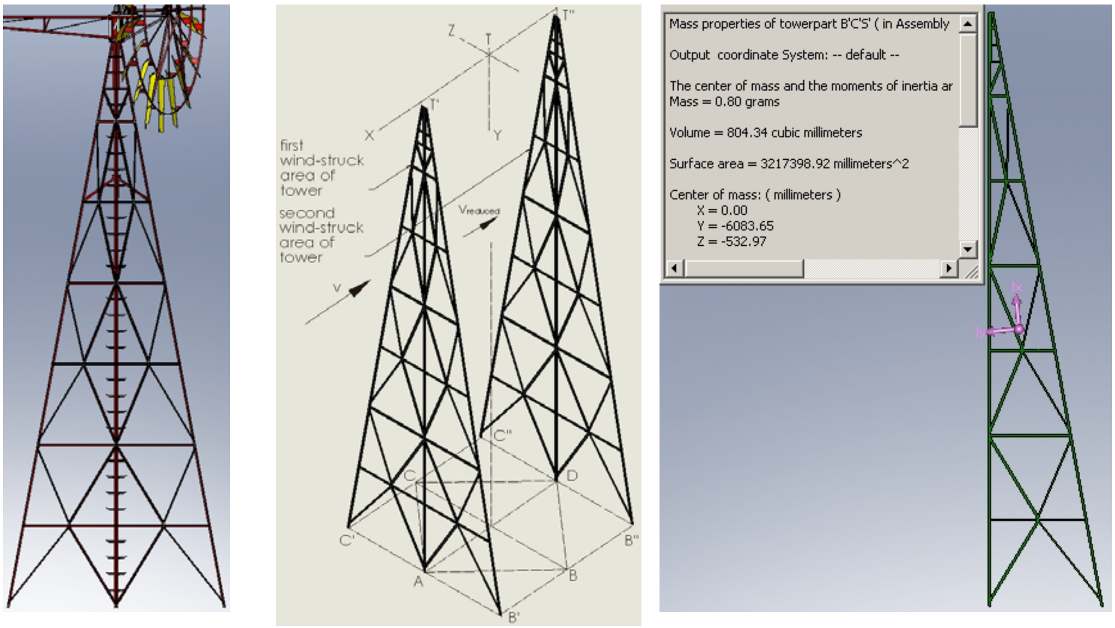
$$Mx_{tower\&etc} = G_{tower\&etc}.Z \approx 6894.0{,}01379 \approx 95Nm$$

$$Mz_{tower\&etc} = G_{tower\&etc}.X \approx 6894.0{,}01809 \approx 125Nm$$

9.2.6 Windforce on tower and parts of rising main

Tower experiences wind over its diagonal. Therefore, tower is split-up in two (parallel) surfaces with, almost, no thickness. Explanation:

- actual tower is situated on base (ABDC) over to top (T)
- first area of tower is at B'C'T', frontally struck with windspeed v
- second area of tower is at B"C"T", frontally struck with windspeed $v_{reduced}$
- $v_{reduced}$ is related to v with a reducing factor ($f_{reduced}$) , set on 0,9
- first - and second area of tower are equal to projected area (perpendicular to winddirection) of tower ($At_{projected} \approx 3{,}22 m^2$, graphically found)



towerpipe partly in wake of rotor	simplified model of tower, two wind-struck areas	area of B'C'T' = area of B"C"T" = $At_{projected}$ = Surface area

Fig. 9.10: tower "seen diagonally" by wind ($v \approx 40 m/s$) and projected area

With simplified model of tower:

- windforce on steps is neglected
- windforce on part of rising main and clamp for rising main is neglected
- windforce on leg BT and leg CT are counted twice
- windforce on part of towerpipe is counted twice

It is assumed, these windforces rule each other out.
Tower is regarded being an angled cube. Drag coefficient (c_d) of an angled cube is approx. 0,80. Windforce on tower and rising main ($F_{t\&rm}$) is:

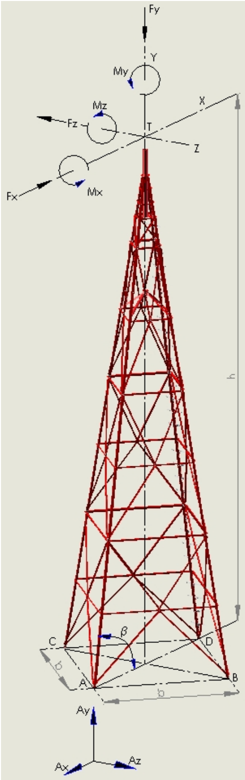
$$F_{t\&rm} = \frac{1}{2}.\rho_{air}.v^2.c_d.At_{projected}.\left(1 + f_{reduced}^2\right) \approx \frac{1}{2}.1{,}225.40^2.0{,}80.3{,}22.\left(1 + 0{,}9^2\right) \approx 4566 N$$

Line of action of this force is parallel to X-axis.
Windforce on tower and rising main ($F_{t\&rm}$) generates a moment around Z-axis:

$$Mz_{t\&rm} = F_{t\&rm}.Y \approx 4566.6{,}08365 \approx 27776 Nm$$

9.2.7 Reaction forces of foundation blocks

When all forces and moments (calculated earlier) are combined, reaction forces of foundation blocks are found.



$$F_x = F_{rotor} + F_{tail} + F_{t\&rm} \approx 3045 + 1042 + 4566 \approx 8653N$$

$$F_y = G_{rhttv} + G_{tower\&etc} \approx 2723 + 6894 \approx 9617N$$

$F_z = 0N$ (Note: free-spinning rotor, because pump is disconnected. Therefore no thrust assumed)

$$M_x = Mx_{gyr} + Mx_{rhttv} + Mx_{tower\&etc} \approx 1409 + 181 + 95 \approx 1685Nm$$

$$M_y = My_{rotor} - My_{tail} + M_{friction} \approx 2628 - 2628 + 191 \approx 191Nm$$

$$M_z = Mz_{gyr} - Mz_{rhttv} + Mz_{tail} + Mz_{tower\&etc} + Mz_{tr\&rm} \Rightarrow$$

$$\Rightarrow M_z \approx 513 - 413 + 208 + 125 + 27776 \approx 28209Nm$$

Height of hub (h) is approx. 10,90m
Base of tower (b) is approx. 2,82m
Angle between leg and horizontal plane (β) is approx. 79,54º.
Reaction forces in A (A_x, A_y, A_z) are shown. Reaction forces in B, C and D are NOT shown, for clarity reasons.

Fig. 9.11: dimensions of tower, with forces, moments and reaction forces

Forces in X-direction:
Assuming each leg consumes equal force: $A_x^F = B_x^F = C_x^F = D_x^F$.

Sum of forces is zero: $F_x - 4.A_x^F = 0 \Rightarrow A_x^F = \frac{F_x}{4} \approx \frac{8653}{4} \approx 2163N$

Forces in Y-direction:
Assuming each leg consumes equal force: $A_y^F = B_y^F = C_y^F = D_y^F$.

Sum of forces is zero: $F_y - 4.A_y^F = 0 \Rightarrow A_y^F = \frac{F_y}{4} \approx \frac{9617}{4} \approx 2404N$

Forces in Z-direction:
Assuming each leg consumes equal force: $A_z^F = B_z^F = C_z^F = D_z^F$.

Sum of forces is zero: $F_z - 4.A_z^F = 0 \Rightarrow A_z^F = \frac{F_z}{4} \approx \frac{0}{4} \approx 0N$

Moments around X-axis:
Assuming Diever 450 tries to tumble around axis AD, then:
$C_y^M = -B_y^M$

Sum of moments is zero: $M_x + F_z.h - 2.C_y^M.\frac{b}{\sqrt{2}} = 0 \Rightarrow C_y^M = (M_x + F_z.h).\frac{\sqrt{2}}{2.b} \Rightarrow$

$$\Rightarrow C_y^M \approx (1685 + 0.10,90).\frac{\sqrt{2}}{2.2,82} \approx 423N$$

Moments around Y-axis:
Assuming each leg consumes equal force: $B_x^M = -A_z^M = -C_x^M = D_z^M$.

Sum of moments is zero: $M_y - 4.B_x^M.\frac{b}{\sqrt{2}} = 0 \Rightarrow B_x^M \approx \frac{191}{4}.\frac{\sqrt{2}}{2{,}82} \approx 24N$.

Note: M_y is solely generated by friction, when yawing. As direction of yawing is unknown, sub-reaction forces ($A_z^M, B_x^M, C_x^M, D_z^M$) need to be taken as positive values.

Moments around Z-axis:
Assuming Diever 450 tries to tumble around axis BC, then:
$D_y^M = -A_y^M$

Sum of moments is zero: $M_z - F_x.h + 2.D_y^M.\frac{b}{\sqrt{2}} = 0 \Rightarrow D_y^M = -(M_z - F_x.h).\frac{\sqrt{2}}{2.b} \Rightarrow$

$$\Rightarrow D_y^M \approx -(28209 - 8653.10{,}90).\frac{\sqrt{2}}{2.2{,}82} \approx 16577N$$

Combining sub-reaction forces:

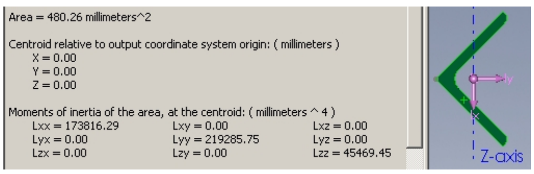
$A_x = A_x^F + A_x^M \approx 2163 + 0 \approx 2163N$	$B_x = B_x^F + B_x^M \approx 2163N + 24 \approx 2187N$
$A_y = A_y^F + A_y^M \approx 2404 - 16577 \approx -14173N$	$B_y = B_y^F + B_y^M \approx 2404 - 423 \approx 1981N$
$A_z = A_z^F + A_z^M \approx 0 + 24 \approx 24N$	$B_z = B_z^F + B_z^M \approx 0 + 0 \approx 0N$
$C_x = C_x^F + C_x^M \approx 2163 + 24 \approx 2187N$	$D_x = D_x^F + D_x^M \approx 2163 + 0 \approx 2163N$
$C_y = C_y^F + C_y^M \approx 2404 + 423 \approx 2827N$	$D_y = D_y^F + D_y^M \approx 2404 + 16577 \approx 18981N$
$C_z = C_z^F + C_z^M \approx 0 + 0 \approx 0N$	$D_z = D_z^F + D_z^M \approx 0 + 24 \approx 24N$

9.2.8 Tower checks

Check on yaw bearings. Upper yaw bearing carries most load. Projected area of upper yaw bearing is taken. Resulting compressive stress (σd) in upper yaw bearing is:

$\sigma d = \frac{\sqrt{Fx_U^2 + Fz_U^2}}{D_{pipe}.t} \approx \frac{\sqrt{8442^2 + 1568^2}}{114{,}3.30} \approx 2{,}5\frac{N}{mm^2}$. For HMPE, this is allowable.

Check on buckling. Legs of tower (Fe 360: $E = 210.10^3 \frac{N}{mm^2}, \sigma_e \approx 240\frac{N}{mm^2}$) are prone to buckling.

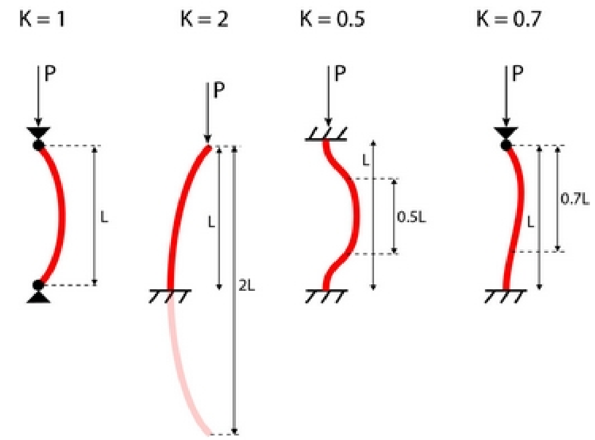


Gyration radius (i_z) is:

$$i_z = \sqrt{\frac{I}{A}} = \sqrt{\frac{L_{ZZ}}{Area}} \approx \sqrt{\frac{4{,}55.10^4}{4{,}80.10^2}} \approx 9{,}73mm$$

Fig. 9.12: towerleg (∟50.50.5), shown with its weakest axis

Factor (K) defines buckling length (l_k), depending on how ends of legs are supported.



It is assumed that:

for welded tower:
$L_w = 1{,}5m$ with $K_w = 0{,}7$

for galvanized tower:
$L_g = 1{,}0m$ with $K_g = 1{,}0$

Fig. 9.13: factor (K), related to way of support

Slenderness (λ) is determined as: $\lambda = \dfrac{l_k}{i_z}$, resulting in:

$$\lambda_{welded} = \frac{L_w.K_w}{i_z} \approx \frac{1{,}5.0{,}7}{9{,}73.10^{-3}} \approx 108 \text{ (for welded tower)}$$

$$\lambda_{galvanized} \approx \frac{L_g.K_g}{i_z} \approx \frac{1{,}0.1{,}0}{9{,}73.10^{-3}} \approx 103 \text{ (for galvanized tower)}$$

$$\lambda_{EULER} = \pi.\sqrt{\frac{E}{\sigma_e}} \approx \pi.\sqrt{\frac{210.10^3}{240}} \approx 93 \text{ (for material Fe 360)}$$

Euler is valid: both towers need to be checked on buckling. Welded tower shows highest slenderness, thus decisive when calculating buckling stress (σ_{buckle}) in legs.

buckling coefficient ω for open profiles (Fe 360)										
λ	ω for λ plus:									
	0	1	2	3	4	5	6	7	8	9
20	1,00	1,01	1,01	1,02	1,03	1,03	1,04	1,05	1,05	1,06
30	1,07	1,08	1,08	1,09	1,01	1,11	1,11	1,12	1,13	1,14
40	1,15	1,15	1,16	1,17	1,18	1,19	1,20	1,21	1,22	1,23
50	1,24	1,25	1,26	1,27	1,28	1,29	1,30	1,31	1,32	1,33
60	1,34	1,35	1,37	1,38	1,39	1,40	1,42	1,43	1,44	1,45
70	1,47	1,48	1,50	1,51	1,53	1,54	1,56	1,57	1,59	1,60
80	1,62	1,64	1,65	1,67	1,69	1,71	1,73	1,75	1,77	1,79
90	1,81	1,83	1,85	1,87	1,89	1,92	1,94	1,97	1,99	2,02
100	2,04	2,07	2,10	2,12	2,15	2,18	2,21	2,25	2,28	2,31
110	2,35	2,38	2,43	2,47	2,51	2,56	2,60	2,65	2,69	2,74
120	2,78	2,83	2,88	2,93	2,97	3,02	3,07	3,12	3,17	3,22
130	3,27	3,32	3,37	3,42	3,47	3,52	3,58	3,63	3,68	3,74
140	3,79	3,84	3,90	3,95	4,01	4,07	4,12	4,18	4,24	4,29
150	4,35									
160	4,95									
170	5,59									
180	6,27									
190	6,98									
200	7,74									

for intermediate values, linear interpolation is allowed

table taken from: Dubbel Taschenbuch für den Maschinenbau
16. Auflage, korrigiert und ergänzt: W. Beitz & K.-H. Küttner
Springer-Verlag Berlin Heidelberg GmbH

Fig. 9.14: buckling coefficient ω for open profiles (Fe 360)

Buckling stress can be calculated, by using a buckling coefficient (ω), which is related to slenderness and can be found in appropriate tables.
For open profiles (like: L-, T-, U-, I- and H-profiles) and made of Fe 360, with slenderness of $\lambda = \lambda_{welded} \approx 108$, buckling coefficient is approx. 2,28 (see table).
Buckling stress (σ_{buckle}) in leg needs to be equal or smaller than yield strength (σ_e):

$$\sigma_{buckle} = \omega.\frac{F}{A} = \omega.\frac{D_y.\frac{1}{\sin\beta}}{Area} \approx 2,28.\frac{18981.\frac{1}{\sin(79,54°)}}{480} \approx 91,7\frac{N}{mm^2}$$

Buckling stress is allowable, because being smaller than yield strength.

Check on connection of leg and foot. Leg is connected to foot with four M10 bolts (class 8.8).

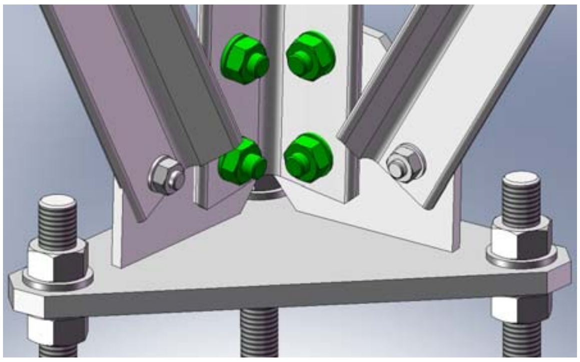


Fig. 9.14: connection between leg and towerfoot

Each M10 bolt is tightened with a torque wrench (tightening torque: 54Nm), resulting in preload ($F_{preload}$) of approx. 28,8kN. Assuming:

- (dry) friction coefficient between leg and foot (f) is approx. 0,3
- forces parallel to leg are solely transmitted by friction
- horizontal reaction forces are taken by tower-diagonals

Hence, maximum generated friction ($W_{\max}$) needs to be equal or more than force parallel to leg:

$$W_{\max} \geq \frac{D_y}{\sin\beta} \Rightarrow 4.f.F_{preload} \geq \frac{D_y}{\sin\beta} \Rightarrow 4.0,3.28800 \geq \frac{18981}{\sin(79,54°)} \Rightarrow 34560N \geq 19302N$$

Equation is true, therefore friction is sufficient to handle forces parallel to leg.

9.2.9 Foundation blocks

Foundation blocks need to be able to deliver reaction forces. It is assumed forces on horizontal plane are absorbed by surrounding soil. Vertical forces are handled by weight of foundation blocks.
When assuming there is no friction between foundation blocks and surrounding soil, and assuming concrete density ($\rho_{concrete}$) is approx. 2200kg/m³ (same assumptions as made earlier), minimum volume (V_{block}) of each foundation block must be:

$$V_{block} = \frac{F}{\rho_{concrete}.g} = \frac{|A_y|}{\rho_{concrete}.g} \approx \frac{|-14173|}{2200.9,807} \approx 0,66m^3$$

As example: when a cube is taken as foundation block, each edge should be minimal 0,87m (≈ 90cm, when rounded up).
Area of base of foundation block (A_{block}) is dependent of soil bearing capacities:

type of soil/ rock	**presumptive bearing capacity (** $BC_{soil/rock}$ **)** [kN/m²]
rock	3240
soft rock	440
coarse sand	440
medium sand	245
fine sand	440
soft shell/ stiff clay	100
soft clay	100
very soft clay*	50

*Note: * can be penetrated several centimetres with the thumb*

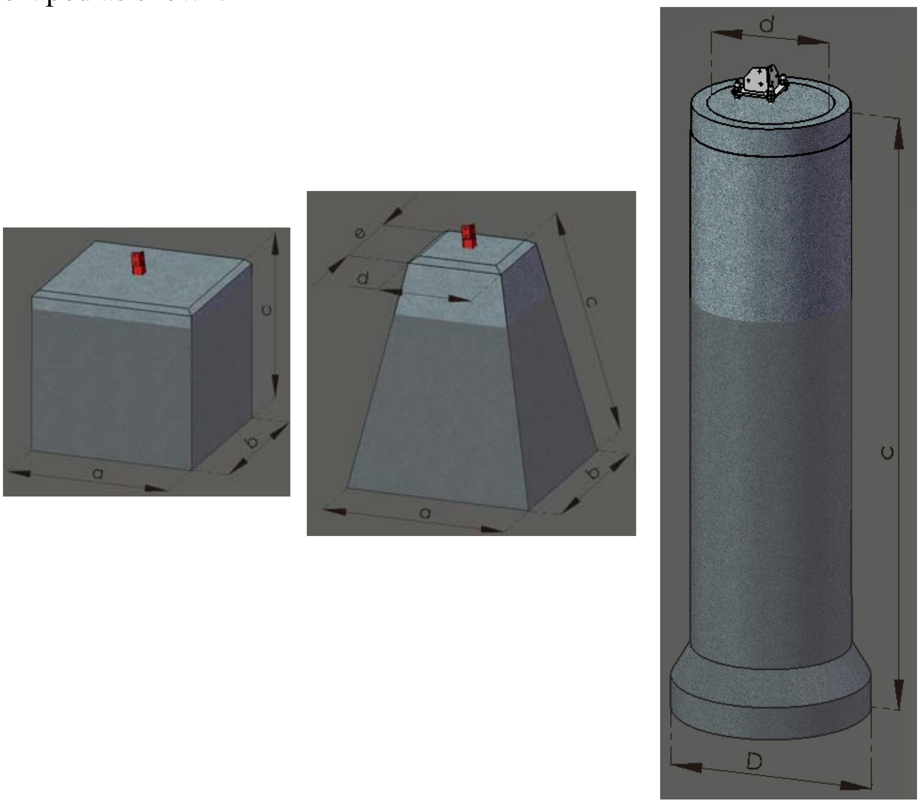
Assuming foundation blocks are supported by bed of soft clay, base of each foundation block (A_{block}) should be minimal:

$$A_{block} = \frac{D_y + |A_y|}{BC_{softclay}} \approx \frac{18981 + |-14173|}{100.10^3} \approx 0,33m^2$$

As example: when a square is taken as base of foundation block, each side should be minimal 0,58m (≈ 60cm, when rounded up). This would result in a block-height of approx. 185cm.

9.3 Types of foundation blocks

Depending on personal preferences and availability of items, foundation blocks can be shaped as shown.



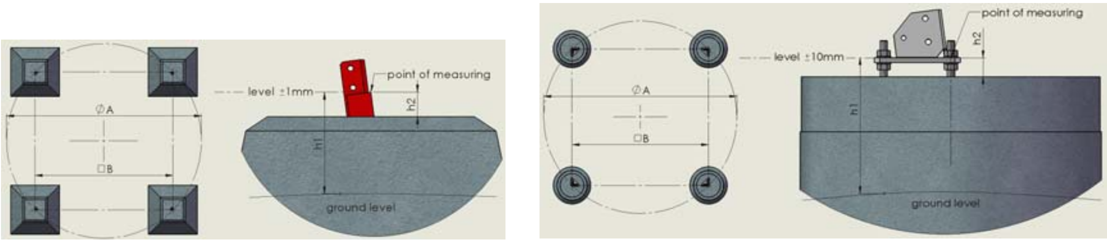
cube	pyramid	socket/ spigot pipe
$a = b = c \approx 0{,}9m$	$a = b \approx 1{,}0m$	$d = 20'' \approx 0{,}5m$
	$c \approx 1{,}2m$	$D \approx 0{,}8m$
	$d = e \approx 0{,}5m$	$c \approx 2{,}4m$
$Area_{base} \approx 0{,}81m^2$	$Area_{base} \approx 1{,}00m^2$	$Area_{base} \approx 0{,}48m^2$
$Volume \approx 0{,}73m^3$	$Volume \approx 0{,}71m^3$	$Volume_{empty} \approx 0{,}31m^3$
		$Volume_{filled} \approx 0{,}82m^3$
$Weight \approx 15728N$	$Weight \approx 15363N$	$Weight_{empty} \approx 7797N$
		$Weight_{filled} \approx 18709N$
$Mass \approx 1605kg$	$Mass \approx 1570kg$	$Mass_{empty} \approx 795kg$
		$Mass_{filled} \approx 1907kg$

Fig. 9.15: types of foundation blocks with size, area, volume, weight and mass

Characteristics of each foundation block:
Cube: vertical sides need shuttering, which is hard too remove. Large top with little height above ground level.
Pyramid: upright sides need shuttering, easy to remove because of draft angle. Small top with some height above ground level.
Socket/ spigot pipe: no shuttering needed. Small top with increased height above ground level, thus increasing gross hub height.

9.4 Spacing

Foundation blocks need to be situated around borehole/ well at correct distances and height. Height of anchors for welded tower (without extensions) is important, as there is no possibility of adjusting height, once concrete has hardened. Anchors should therefore be water-levelled, facilitating tower standing vertical. For galvanized tower, or when extensions are used under welded tower, feet can be adjusted with nuts on stud bolts, so whole tower can be set exactly vertical after concrete has hardened.



welded tower without extensions

welded tower with extensions and galvanized tower with/without extensions

Fig. 9.16: foundation set-up

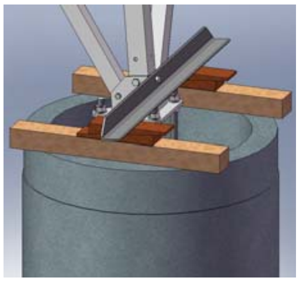
Foundation distances are influenced by combinations of type of tower, type of foundation block and which section is used as lowest tower-section. For ease of excavating holes for foundation blocks, some configurations are given in table below.

section 3/4/5/6/7 & hub height ≈ [m]		tower type	foundation block type	□B [mm]	h1 ≈ [mm]	h2 [mm]	ØA ≈ [mm]
3	8,0	welded	cube	2050	100	50	2899
4*	10,9	welded	pyramid	2820	300	50	3988
5	13,9	welded	socket/ spigot pipe	3620	800	50	5119
6	16,8	welded	socket/ spigot pipe	4390	800	50	6208
7	19,8	welded	socket/ spigot pipe	5160	800	50	7297
3	7,6	galvanized	pyramid	2080	300	50	2942
4**	10,6	galvanized	socket/ spigot pipe	2850	800	50	4031
5	13,5	galvanized	socket/ spigot pipe	3620	800	50	5119
6	16,5	galvanized	socket/ spigot pipe	4390	800	50	6208
7	19,4	galvanized	socket/ spigot pipe	5160	800	50	7297
* = standard welded tower, ** = standard galvanized tower							

Fig. 9.17: foundation distances with regard to tower type, foundation block type and lowest tower-section used

It is recommended to assemble lowest section of tower (with horizontals attached to top) and place them correct in height (h2), horizontal position (with regard to borehole/ well) and water-levelled. For this, four short angles, bolted to bottom of lowest section, going across to shuttering, and together with some wedges, are helpful (see fig. 9.18). Attach anchors or feet with stud bolts to lowest section of tower. When satisfied, concrete can be poured. Note that top of the concrete has to slope down outwards from the anchor/ stud bolts to prevent water collection.

welded tower with anchor



galvanized tower with foot

Fig. 9.18: examples of supporting lowest section of tower, prior to concrete pouring

10 Hoisting, a variation of installation

10.1 General

This report described installation of rotor, transmission and tail with use of jib and chain hoist, when tower is upright. In that case, tower has to be erected with a (mobile) crane, or has to built up gradually, using a jib. This procedure is valid, but time-consuming, because mounting activities have to be done on height.
For that reason, hinged feet for tower can be provided. Facilitating assembling (parts of) Diever 450 horizontally and when done, hoisting it to vertical. This action needs to have a stable anchor point away from location of Diever 450 and a wire rope hoist (also called: tirfor (from French: tir = pull and for(t) = force)) to complete the job. Foundation blocks of windmill need to be aligned with anchor point for tirfor.

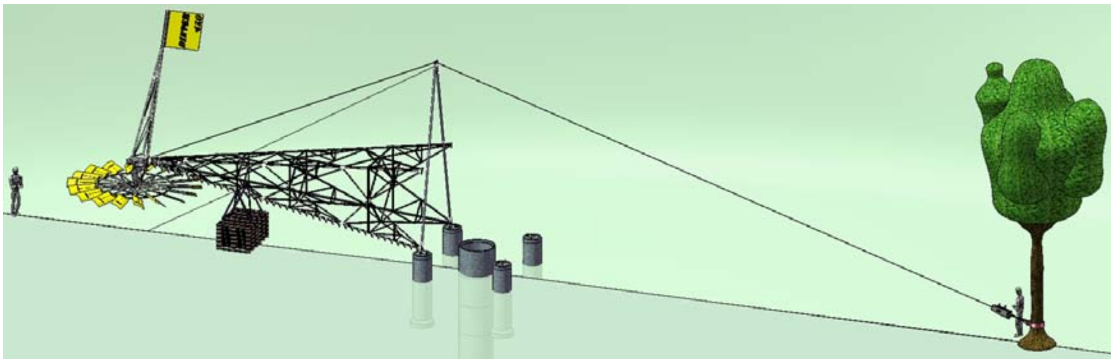
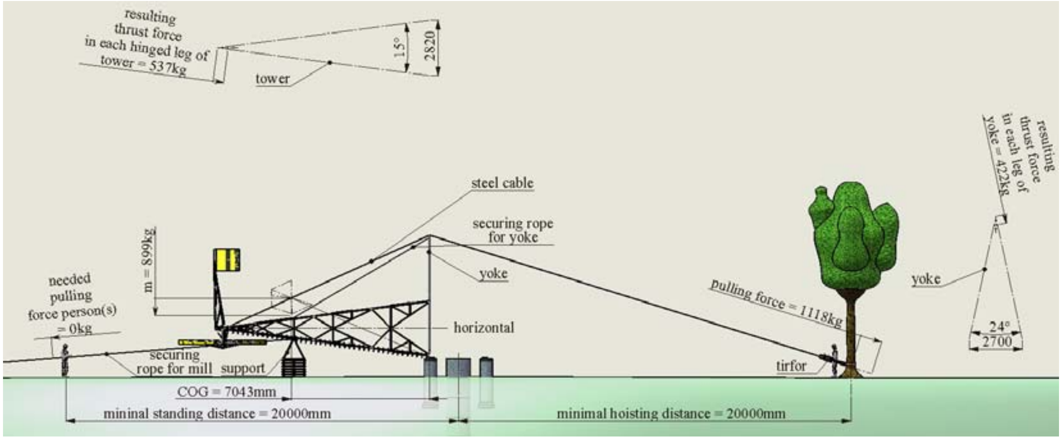
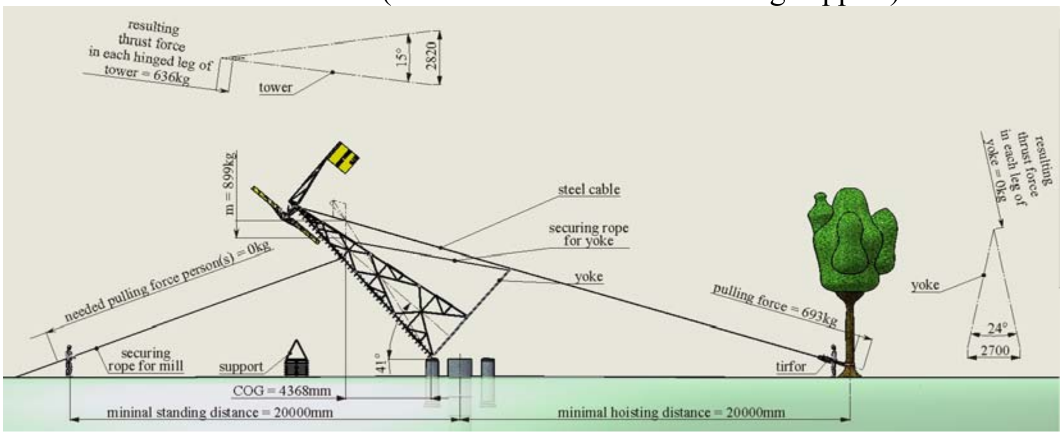


Fig. 10.1: Diever 450 (galvanized) assembled horizontal; direction of foundation blocks, anchor point for tirfor, yoke, steel cable, support and securing ropes

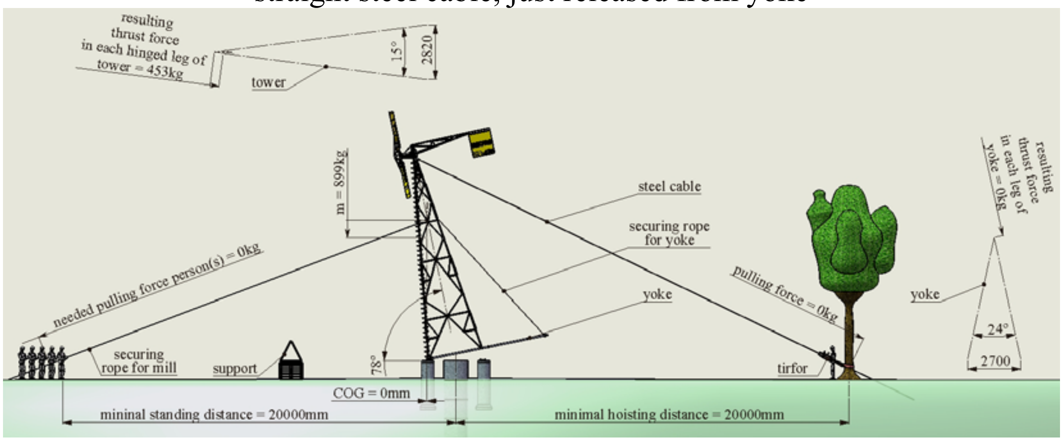
A yoke is needed for first part of hoisting, when tower still is more or less horizontal. It supports a steel cable. When reaching a certain angle during hoisting (see fig. 10.2), steel cable will become straight and is released from yoke. Yoke is hinged with regard to tower and therefore can be mounted when horizontal. When done, yoke and steel cable are rotated upwards and fixated as shown. Securing ropes are attached. Transmission is locked by rotation limiter. Before hoisting, fasteners for feet should be greased and tools placed, ready to pick, to provide swift mounting when Diever 450 is vertical. A safe distance should be kept (at all time!) during hoisting.



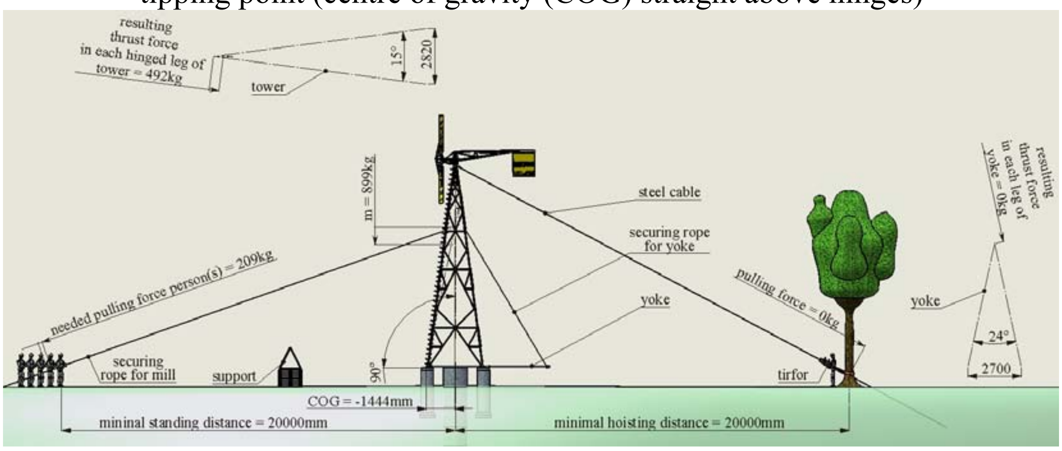
tower horizontal (assumed: tower NOT touching support)



straight steel cable, just released from yoke



tipping point (centre of gravity (COG) straight above hinges)



tower vertical (assumed: two towerfeet NOT touching foundation plates)

Fig. 10.2: hoisting, shown at interesting angles, indication of forces [kg]

10.2 Yoke

Yoke is, most likely, used only once. For that reason, it is chosen to use 2” gaspipes which can be re-used as rising main (described in chapter 11), once windmill is upright. If larger rising main is used, yoke can be made accordingly. Threaded gaspipe is bolted (using flanges) between brackets. Length of each leg of yoke is set on six meter. Bottom brackets are bolted with M16 to feet of tower. Top bracket is fitted with a 2” long sweep bend to provide smooth guidance for steel cable. On both sides of top bracket, shackles are mounted to facilitate tying of ropes for securing. Weight approx. 75kg.

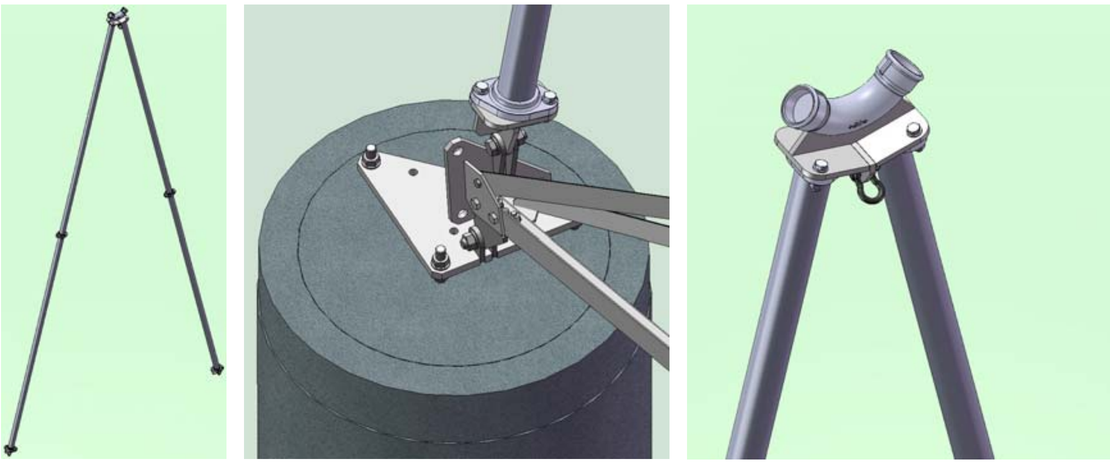
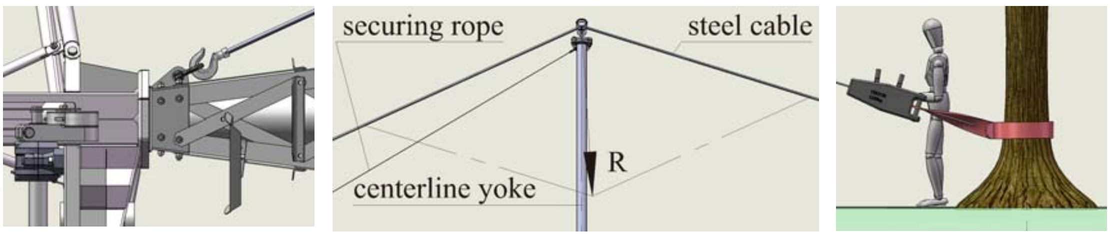


Fig. 10.3: yoke, connection of bottom bracket with towerfoot, top bracket

Steel cable is attached at top of tower, over to top bracket of yoke and entering tirfor at anchor point.



towertop (horizontal) yoke (vertical) tirfor at anchor point

Fig. 10.4: attachments of steel cable

When tower is horizontal and yoke is vertical, resultant of force in steel cable (R) should pass centreline of yoke at right as shown in fig. 10.4, to induce a small tension force in securing rope. In that way, theoretical minimal hoisting distance can be deduced easily, as being the height of tower. For safety, minimal hoisting distance for 10,5m tower is set on 20 meters (as shown in pictures). During hoisting, angle between resultant and centreline of yoke will become larger. As a result, securing rope will get more tensioned, which is regarded as positive. When tower has reached an angle where steel cable has become straight, securing rope will prevent yoke from falling.

10.3 Concise calculations

Forces at various hoisting angles are given:

hoisting angle [º]	**pulling force (tirfor)** [kg]	**pulling force person(s)** [kg]	**thrust force in each hinged leg of tower** [kg]	**thrust force in each leg of yoke** [kg]
-10	1140	0	468	495
-7,4*	1138	0	488	479
-5	1133	0	505	462
0	1118	0	537	422
5	1094	0	566	376
10	1061	0	590	325
15	1021	0	610	271
20	972	0	625	215
25	917	0	636	159
30	854	0	641	105
35	785	0	642	54
40	709	0	638	8
41,0**	693	0	636	0
45	585	0	602	0
50	467	0	567	0
55	363	0	538	0
60	271	0	514	0
65	188	0	494	0
70	113	0	477	0
75	44	0	462	0
78,4***	0	0	453	0
80	0	27	459	0
85	0	115	475	0
90	0	209	492	0

Fig. 10.5: forces [kg] when hoisting Diever 450 (galvanized included: tower, head, rotor & transmission) with yoke of 6 meters and hoisting distance of 20 meters at various hoisting angles
* = hinged legs of tower horizontal
** = straight steel cable, just released from yoke
*** = tipping point (centre of gravity (COG) straight above towerhinges)

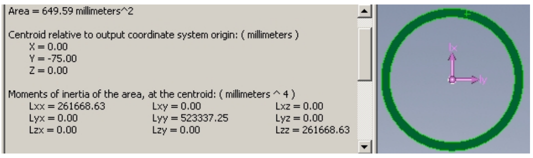
Check on hinged legs of tower. During hoisting, maximum thrust force on each hinged leg of tower will occur at 35 º from horizontal (fig. 10.5). Check on buckling stress in hinged towerlegs is similar to calculations done in chapter 9.2.8. In this case, $\lambda_{galvanized} \approx 103$, resulting in buckling coefficient of $\omega \approx 2{,}12$.

Buckling stress (σ_{buckle}) in leg needs to be equal or smaller than yield strength (σ_e):

$$\sigma_{buckle} = \omega.\frac{F}{A} = \omega.\frac{m.g}{Area} \approx 2{,}12.\frac{642.9{,}807}{480} \approx 27{,}8\frac{N}{mm^2}$$

Buckling stress in hinged towerlegs is allowable, because being smaller than yield strength.

Check on legs of yoke. During hoisting, maximum thrust force on each leg of yoke occurs at -10 º (fig. 10.5). With this value of thrust force is being calculated, although in most cases, negative angles will not be needed. In event Diever 450 is installed on a slope, it is advised to assemble windmill with head "uphill" to ease hoisting activities. Length of each leg of yoke(L) is set on six meter and is made from 2" gaspipe.



Gyration radius (i) is:

$$i = \sqrt{\frac{I}{A}} = \sqrt{\frac{L_{ZZ}}{Area}} \approx \sqrt{\frac{26{,}17.10^4}{6{,}50.10^2}} \approx 20{,}07mm$$

Fig. 10.6: leg of yoke (gaspipe 2"), shown with moment of inertia and area

Gyration radius of a pipe can also be calculated (with diameters taken from fig. 11.2):

$$i = \frac{1}{4}.\sqrt{\frac{D^4 - d^4}{D^2 - d^2}} = \frac{1}{4}.\sqrt{\frac{D_{outer}^4 - D_{rm}^4}{D_{outer}^2 - D_{rm}^2}} \approx \frac{1}{4}.\sqrt{\frac{60{,}3^4 - 53{,}0^4}{60{,}3^2 - 53{,}0^2}} \approx 20{,}07mm$$

, logically leading to same result as in fig. 10.6.

It is assumed, brackets are welded at correct incline with regard to yoke. For that case, factor (K), which is related to way of support (fig. 9.13) is taken 0,7.

Slenderness (λ) is determined as: $\lambda = \frac{l_k}{i}$, resulting in:

$$\lambda_{yoke} = \frac{L.K}{i} \approx \frac{6{,}0.0{,}7}{20{,}07.10^{-3}} \approx 209$$

buckling coefficient ω for hot finished hollow sections (Fe 360)										
λ	ω for λ plus:									
	0	1	2	3	4	5	6	7	8	9
20	1,00					1,01				
30	1,02					1,02				
40	1,05					1,07				
50	1,09					1,12				
60	1,16					1,21				
70	1,26					1,32				
80	1,39					1,49				
90	1,60					1,74				
100	1,93					2,13				
110	2,33					2,55				
120	2,78					3,02				
130	3,26					3,52				
140	3,78					4,06				
150	4,34					4,64				
160	4,94					5,25				
170	5,58					5,91				
180	6,25					6,61				
190	6,96					7,34				
200	7,72									

for intermediate values, linear interpolation is allowed

table taken from: Dubbel Taschenbuch für den Maschinenbau 16. Auflage, korrigiert und ergänzt: W. Beitz & K.-H. Küttner Springer-Verlag Berlin Heidelberg GmbH

Fig. 10.7: buckling coefficient ω for hot finished hollow sections (Fe 360)

For hot finished hollow sections like pipes and rectangular profiles, buckling coefficient (ω) can be found in fig. 10.7.
Note: table for hot finished hollow sections shows, these profiles are slightly more resilient to buckling when compared with open profiles (table shown in fig. 9.14).

Although officially not allowed, buckling coefficient for yoke (with $\lambda = 209$) is found by linear extrapolation, using nearest given values for ω (at $\lambda = 195$ and at $\lambda = 200$).

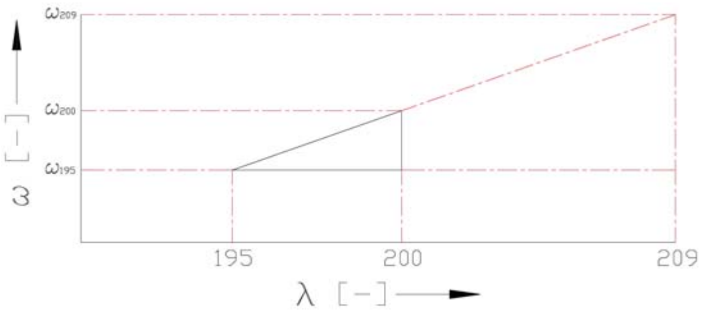


Fig. 10.8: buckling coefficient ω at $\lambda = 209$ (ω_{209}), found by linear extrapolation

This results in:

$$\frac{\omega_{209} - \omega_{195}}{209 - 195} = \frac{\omega_{200} - \omega_{195}}{200 - 195} \Rightarrow \omega_{209} = \frac{(209 - 195).(\omega_{200} - \omega_{195})}{200 - 195} + \omega_{195} \Rightarrow$$

$$\omega_{209} = \frac{(209 - 195).(7{,}72 - 7{,}34)}{200 - 195} + 7{,}34 \approx 8{,}40$$

Buckling stress (σ_{buckle}) in legs of yoke needs to be equal or smaller than yield strength (σ_e):

$$\sigma_{buckle} = \omega.\frac{F}{A} = \omega_{209}.\frac{m.g}{Area} \approx 8{,}40.\frac{495.9{,}807}{650} \approx 62{,}7\frac{N}{mm^2}$$

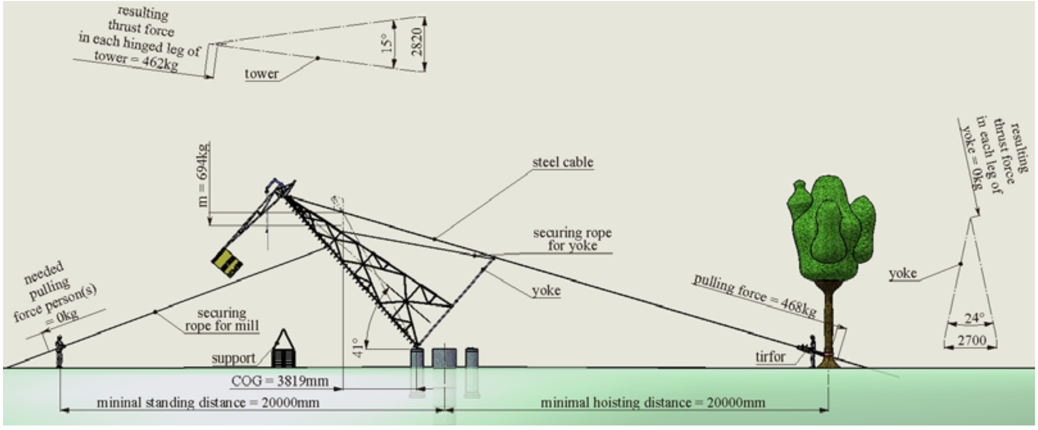
Buckling stress in legs of yoke is allowable, because being smaller than yield strength.

10.4 Hoisting without rotor

Tail has to be fitted to head (see fig. 10.2) before rotor can be mounted. For being able to fit rotor (with main bearings, crank and conrod), it is necessary to rotate head, so tail is approx. 45º from horizontal, facing up. Setting can be kept by using ropes founded to surrounding soil. Some acrobatic feat is needed when rotor is fitted and also, connection between conrod and tumbler is not easy to be made, since rotor is restricting easy access.
When windmill is upright, rotor immediately will direct itself facing wind. Depending on winddirection, rotor may get tangled up with steel cable. Therefore, towerhinges need to be fastened to foundation plates as fast as possible, to be able to release steel cable.

To avoid these drawbacks, a solution to be considered is, to hoist windmill without rotor. In that case, jib and chain hoist should be mounted before hoisting of tower, because once tower is upright, it is hard to fit jib and chain hoist. Rotor is hoisted when tower is vertical, as shown in fig. 0.2. As a bonus, main bearings of rotor will not get exposed to axial load when hoisting. When rotor has been mounted, jib can be easily removed by using a pulley, which is tied to a spoke-ring connection of rotor.

Pulley (with rope) can be fitted when standing on towersteps. Then rotor is turned manually 180º, so that transmission can be locked (with rotation limiter) and so, resulting in a pulley above the jib. Jib can be used if intensive maintenance of Diever 450 is required and therefore should be stored in a secure place, when not in use.



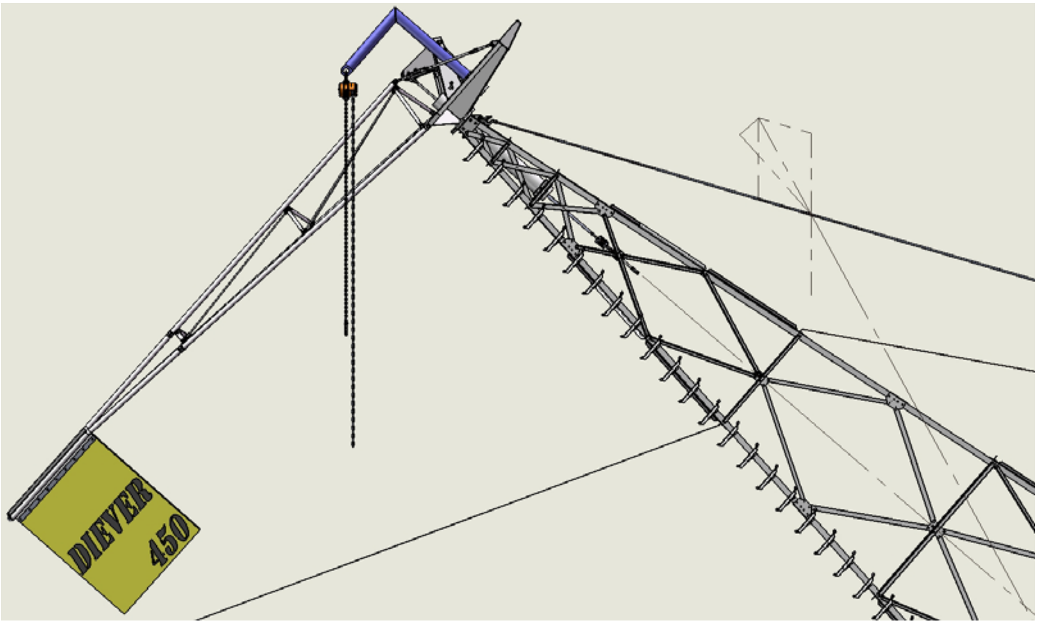


Fig. 10.9: hoisting Diever 450 (galvanized), without rotor, but with jib, chain hoist, head, tail, tumbler, (part of) pumprods, load limiter and shock absorber

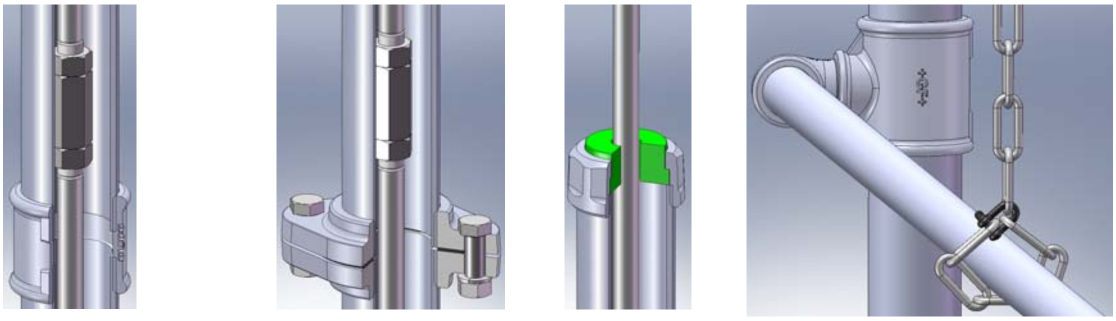
11 Miscellaneous

11.1 Rising main

There are three functions of rising main:

- Fixating pump at right depth in borehole/ well
- Guiding of pumprod, in order to prevent buckling at high windspeeds
- Transporting water, delivered by pump, to storagetank

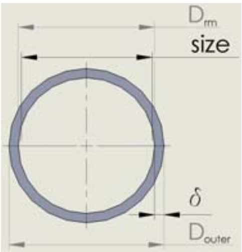
Normally, galvanized gaspipe is chosen for rising main, because they are delivered with British Standard Tapered Pipe Thread (BSPT) (R), to facilitate sealing gaps (with teflon-tape or hemp) at joints. For Diever 450, rising main of minimal 2” gaspipe is required. Larger is better, especially for low pumping heads, to limit peakforces in pumprod. For extractable pumps, inner diameter of rising main is important. Only when size of rising main (gaspipe) is chosen in accordance of size of extractable pump, properties of extractable pumps can be fully exploited.



rising main with socket, pumprod-connection at LDC | rising main with flanges, pumprod-connection at LDC | pumprod guide | delivery pipe to storagetank

Fig. 11.1: rising main, connections and other parts

It is recommended to limit pipe length to 3 meter, because of weight handling. It will also ease bringing them into the tower. Indication of size and weight of gaspipes:



size	δ [mm]	$D_{outer} \approx$ [mm]	$D_{rm} \approx$ * [mm]	*weight* $\approx$ ** [kg]
1½"	3,25	48,3	41,8	10,8
2"	3,65	60,3	53,0	15,3
2½"	3,65	76,1	68,8	19,6
3"	4,05	88,9	80,8	25,4

* D_{rm} is inner diameter of pipe ** *weight* is based on pipe length of three meter

Fig. 11.2: examples of gaspipe

Use flanges for joining rising main, instead of sockets (if possible, with regard to inner diameter of borehole). In that way, thread on pipe will not get damaged during transport and rising main is easily (dis)assembled on site.

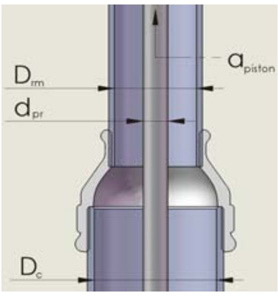
At top of rising main a pumprod guide (made of HMPE) is mounted, to prevent items falling into the rising main, which would otherwise end up in the pump. Pumprod guide is not watertight.

Fig. 11.3: pumprod guide and clamp for rising main (photo 2018)

Depending on height of storagetank, a delivery pipe exits rising main. For Diever 450, maximum height of delivery pipe is approx. 2,7 meter below hub height.

11.2 Pumprods

Function of pumprods is to transmit power from transmission to piston of reciprocating pump. As a result, force in pumprod will fluctuate during each revolution of rotor. When transmission was evaluated in 2004 [lit. 9], it was assumed highest force in pumprod will occur just after LDC, when piston-valve has closed. At that moment, water in rising main, still resting on foot-valve, bumps against moving and accelerating piston. Resulting in a peak force in pumprod. For this reason transmission was re-designed, lowering piston accelerations during upward stroke. CWD (Consultancy Services Wind Energy Developing Countries) introduced an empirical formula to estimate peak force (F_{peak}) in pumprod:



$$F_{peak} \approx k.\rho_{water}.\left(g + a_{piston}.\frac{D_c^2 - d_{pr}^2}{D_{rm}^2 - d_{pr}^2}\right).H.\frac{\pi}{4}.\left(D_c^2 - d_{pr}^2\right)$$

With this formula, dynamic waterpressure, developed by acceleration of water in rising main is added to static waterpressure in pump. Result is multiplied by effective area of piston, thus obtaining peakforce in pumprod.

Fig. 11.4: pump, rising main, pumprod, acceleration of piston

In which:

- k is overshootfactor, from experiences between 1,5 and 2,0
- a_{piston} is acceleration of piston at peak-moment [m/s²]
- D_c is (inner) diameter of pump-cylinder (as calculated in chapter 8.2) [m]
- d_{pr} is diameter of pumprod (to be chosen) [m]
- D_{rm} is (inner) diameter of rising main (see fig. 11.2) [m]
- H is pumping head of Diever 450 [m]

When diameter of pumprod is neglected, formula can be simplified: $F_{peak} \approx \frac{A_0}{H} + B_0$

in which: $A_0 = k.\rho_{water}.a_{piston}.\frac{\pi}{4}.\frac{\Delta_{450}^4.v_{mean}^4}{D_{rm}^2}$ and $B_0 = k.\rho_{water}.g.\frac{\pi}{4}.\Delta_{450}^2.v_{mean}^2$

Note: A_0 & B_0 are parameters, related to overshootfactor, mean (annual) windspeed, rotor speed and size of rising main. Formula is only valid, when correct size pump has been installed with regard to pumping head and mean (annual) windspeed. Acceleration of piston is equal to acceleration of pumprod. Fig. 5.3 shows, during upward stroke, maximum acceleration of pumprod is: $a_{up\_60rpm} \approx 2,830\frac{m}{s^2}$ at rotor speed 60rpm and $a_{up\_80rpm} \approx 5,031\frac{m}{s^2}$ at rotor speed 80rpm. CWD formula was used to calculate pumprod forces (at 12m and at 36m pumping head) with pumprods of Ø10mm, Ø12mm and Ø16mm to obtain relevant A_c and B_c. Assumed in all calculations: overshootfactor $k = 2$ and 2” gaspipe as rising main. Results show diameter change of pumprod does NOT significantly impact peak force in pumprod. So, in most cases, size of pumprod can be neglected, when calculating peak force.

Values of parameters A and B are given:

$F_{peak} \approx \frac{A}{H} + B$		valid for all pumprods, gives approx. values for F_{peak}		valid for relevant pumprod diameters, gives more exact values for F_{peak}					
pumprod size [mm]:		Ø0 – Ø16		Ø10		Ø12		Ø16	
v_{mean} [m/s]	rotor speed [rpm]	A_0 [Nm]	B_0 [N]	A_{10} [Nm]	B_{10} [N]	A_{12} [Nm]	B_{12} [N]	A_{16} [Nm]	B_{16} [N]
2,5	60	15092	1504	16308	1399	16851	1353	18260	1233
2,5	80	26830	1504	28473	1375	29211	1317	31136	1170
3,0	60	31295	2166	33109	2047	33929	1994	36087	1857
3,0	80	55634	2166	58341	2012	59572	1943	62829	1764
3,5	60	57978	2948	60776	2812	62054	2752	65446	2594
3,5	80	103070	2948	107527	2764	109570	2681	115021	2466

For pumprod of Ø12mm, peak forces, at rotor speeds (60rpm & 80rpm) are put in a graph (see fig. 11.5).
For pumprods Ø10mm, Ø12mm and Ø16mm, elongations of pumprods during peak, when rotor speed is 80rpm, are put in a graph (see fig. 11.6).

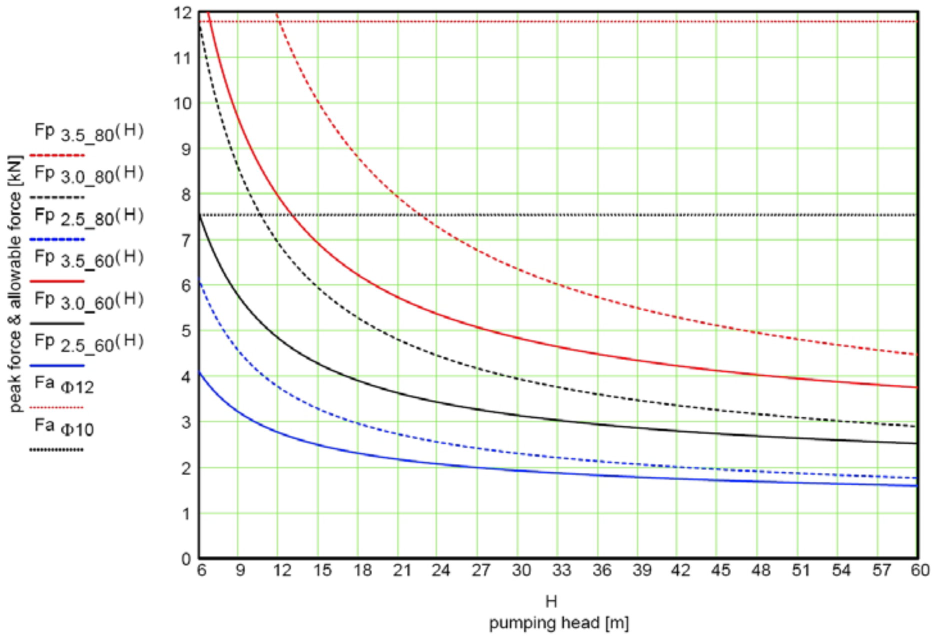


Fig. 11.5:

- peak force in pumprod of Ø12mm, when correct size pump has been installed with regard to pumping head and mean (annual) windspeed (2,5m/s, 3,0m/s or 3,5m/s), installed with rising main of 2” gaspipe, at two rotational speeds of rotor (60rpm and 80rpm), shown at various pumping heads
- allowable (static) force of pumprod Ø10mm and Ø12mm

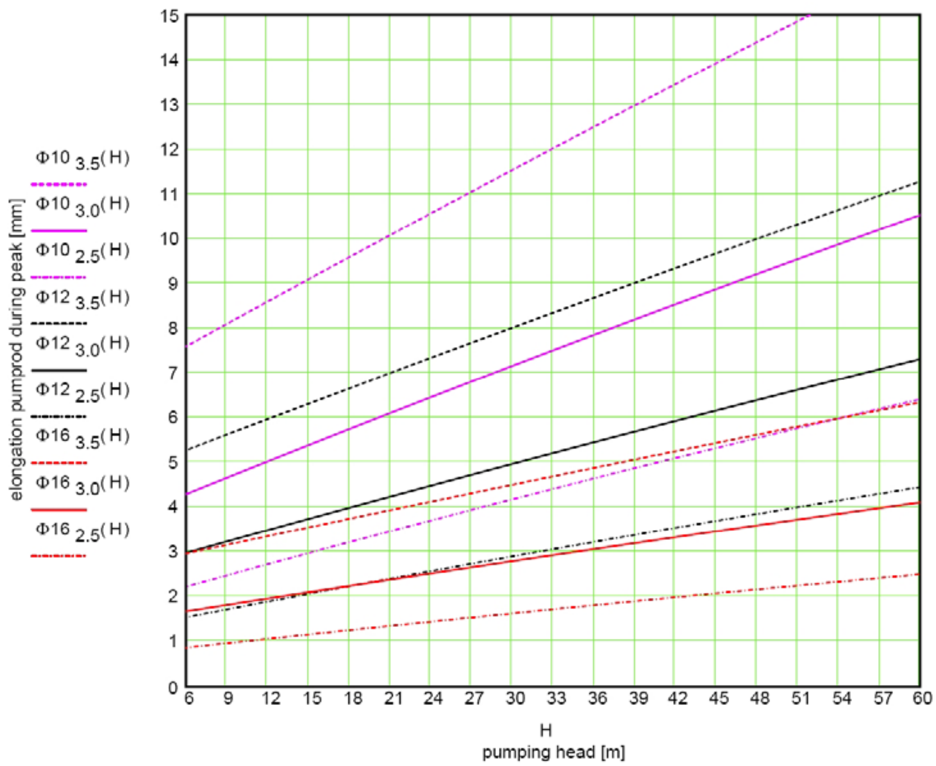


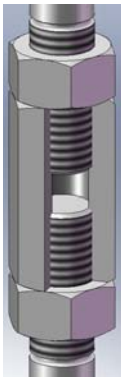
Fig. 11.6: elongation of pumprod (Ø10mm, Ø12mm and Ø16mm) during peak, when correct size pump has been installed with regard to pumping head and mean (annual) windspeed (2,5m/s, 3,0m/s or 3,5m/s), when installed with rising main of 2” gaspipe, at rotational speed of rotor 80rpm, shown at various pumping heads

In 1993 Diever 450 on WOT testfield (with a pumping head of approx. 12m) was furnished with pumprods Ø16mm (availability reason being).

According to lit. 4, allowable (static) pumprod force for pumps is approx. 10kN. Same report shows duration of peak force is short. Peak force is smoothed by elongation of pumprod. Using smaller diameter of pumprod increases elongation.
Fig. 11.5 and fig. 11.6 show highest peak forces occur at low pumping heads, when elongation (because of relatively short pumprod) is minimal. With other words: peak force is barely smoothed by elasticity of pumprod at low pumping heads.

Concluding:

- For low pumping heads ($H < 12$m). Peak force is high. Larger rising mains (for example 2½" or 3") should be considered. A shock absorber (discussed later in this report) is useful. Pumprods Ø12mm.
- For average pumping heads (12m $< H <$ 24m). Peak force is medium. Rising main 2". Shock absorber might be useful. Pumprods Ø12mm.
- For high pumping heads ($H >$ 24m). Peak force is low. Rising main 2". No need for shock absorber. It is not interesting to smoothen peak forces by reducing pumprod diameter further. Long pumprod provides enough elongation. Pumprods Ø12mm.



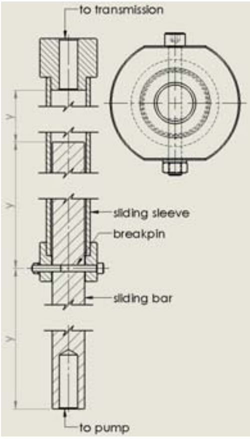
Pumprod connections are made with a coupling nut (h = 3 x d, DIN6334) which is jammed by two “normal” nuts (DIN934), providing for length adjustment of pumprods. Ideally, thread on pumprod is made on a lathe, ensuring straight threads on both ends. End of thread can be fitted with a round groove to minimize risk of fatigue.

Fig. 11.7: pumprod connection, coupling nut jammed by nuts, pumprod with groove

Pumprods should have same length as rising mains. In that way every connection of pumprod is near to connection of rising main. To provide easy assembling, each pumprod connection should be above rising main connections, when at LDC (see fig. 11.1). Weight of pumprod Ø12mm, with length of three meter is approx. 2,7kg (for comparison: weight of same length pumprod Ø16mm is approx. 4,7kg).

11.3 Load limiter

Function of load limiter is to disconnect pump from transmission when maximum allowable pumprod force is exceeded. Pumprod force is dependent of rotor speed and pump. Overloading can happen when rotor speed gets too high (at severe storm conditions) or when pump gets jammed (for whatever reason). Main goal is to prevent (further) damage to pump, pumprods or transmission, simply by switching off the load, in case of emergency.
When investigating the cause of disconnection, the opportunity can be taken to check for loose bolts and to re-grease bearings of transmission. The connection of load limiter must be restored manually.
In 1993 it was chosen to place load limiter in the part of pumprods where there is no rising main (between towerpipe and rising main clamp, see fig. 3.2). Design of load limiter was copied from renomated windpump manufacturer Southern Cross (Australia), which uses a piece of tropical hardwood as load limiter. According to some literature, wood is less prone to fatigue than steel. Medio 2003, pump of Diever 450 at testfield got jammed (at half stroke position) and as a result the wooden beam snapped. Therefore rotor could rotate freely. Unfortunately, top part of the wooden beam started hammering on lower part, resulting in buckled pumprod-parts. It was decided to improve load limiter to prevent undesired damage in the future.



cross-section

photo 2018

Fig. 11.8: load limiter

Fig. 11.8 shows a cross-section of load limiter as used on Diever 450 at testfield since 2004. A sliding sleeve is connected (by means of pumprods) to the transmission and a sliding bar is attached (also by means of pumprods) to the pump. Joint between the two is made by using a breakpin. Breakpin is in double shear and will snap in case of overload or fatigue. There are four scenarios:

- rotor speed is too high, breakpin snaps, piston of pump falls
- piston of pump gets jammed in LDC, breakpin snaps
- piston of pump gets jammed in TDC, breakpin snaps
- breakpin snaps because of fatigue, piston of pump falls

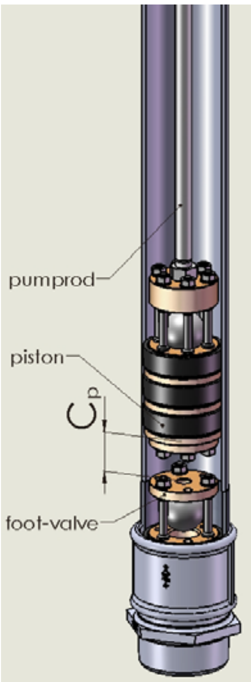
In any scenario, the sliding bar must stay inside the sliding sleeve to prevent hammering. Distances (y) must be more than the stroke (250mm). In that way, when breakpin has snapped, the sliding sleeve has freedom to move up and down while constantly embracing the sliding bar.

In 2004 [lit. 9] diameter of breakpin, in relation to allowable (static) pumprod force for pumps (10kN) [lit. 4], was determined at Ø5mm (resulting in a maximum pumprod force of approx. 8,170kN). According to fig. 11.5 allowable (static) force of pumprod Ø12mm is approx. 11,781kN. Load limiter protects pump ánd pumprods. Sliding sleeve is made of gaspipe 1”, which is welded at both ends to bushes. Sliding bar is Ø25mm. For breakpin, a socket head cap screw M5 x 50 (class 8.8) is used. Distance y is set on 320mm.

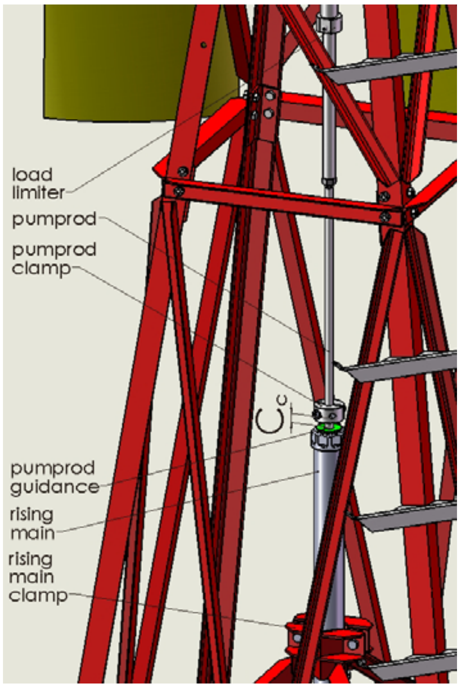
Load limiter, as described, proved reliable. Despite small size of breakpin (and therefore, occasionally snapping, due to fatigue), it is advised NOT to ream the hole to fit a breakpin Ø6mm, as that would lead to maximum pumprod force of 11,753kN. Being more than allowable (static) pumprod force for pumps. Weight: approx. 4,7kg.

11.4 Pumprod clamp

Function of pumprod clamp is to protect pump-parts against damage when breakpin of load limiter snaps. During pumping cyclus, when transmission reaches LDC, there is vertical clearance (inside the pump) between piston and footvalve (C_p):



pump (C_p)



pumprod clamp (C_c)

Fig. 11.9: clearances at LDC

In two scenarios (mentioned in chapter 11.3), when breakpin snaps, piston of pump and most pumprods will fall. Their fall must be limited to avoid damage to piston or foot-valve. For this reason, a pumprod clamp is tightened around pumprod, above pumprod guidance with clearance (C_c), as shown.

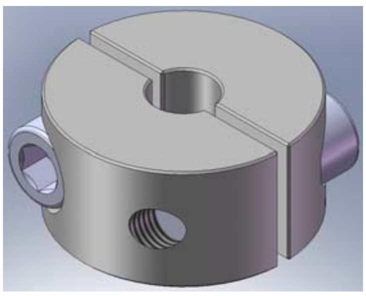


Fig. 11.10: pumprod clamp with socket head cap screw M12 (photo 2018)

Pumprod clamp is adjustable in height and should be fitted so, that: $C_p > C_c$. In that way, pumprod clamp will touch pumprod guidance before falling piston can hit foot-valve, in case of snapping of breakpin.

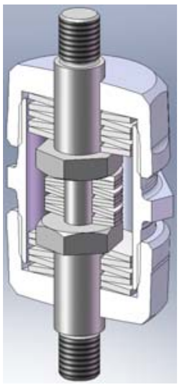
If clearances of $C_p = 80mm$ and $C_c = 30mm$ are wished for, procedure is:

- Transmission is locked at half stroke position (see fig 5.2) with help of rotation limiter
- Load limiter is connected with transmission
- Rising main clamp is fitted, but NOT tightened
- When last pipe of rising main (with pumprod) is fitted, structure is hoisted (with chain hoist) so, connection between pumprod and load limiter can be made
- Pumprod is connected to transmission
- Rising main is lowered (with chain hoist) with distance Y being,
  $Y = \frac{stroke}{2} + C_p = \frac{250}{2} + 80 = 205mm$
- Rising main clamp is tightened
- Rising main is set on correct height: clearance of pump is OK
- Pumprod clamp is tightened with clearance above pumprod guidance, $C_c = 30mm$
- Pumprod clamp is set on correct height, clearance of pumprod clamp is OK
- Transmission is unlocked and rotation limiter is stored
- Check clearance C_c, when transmission is at LDC
- Chain hoist is stored

Clearance of pumprod clamp should NOT be chosen smaller than approx. 30mm to prevent body-parts (like thumbs) getting jammed, when transmission is unlocked. Clearance of pumprod clamp should NOT exceed $y - stroke = 320 - 250 = 70mm$ to prevent hammering of load limiter parts, in case piston of pump and most pumprods fall when breakpin has snapped. Weight: approx. 0,5kg.

11.5 Shock absorber

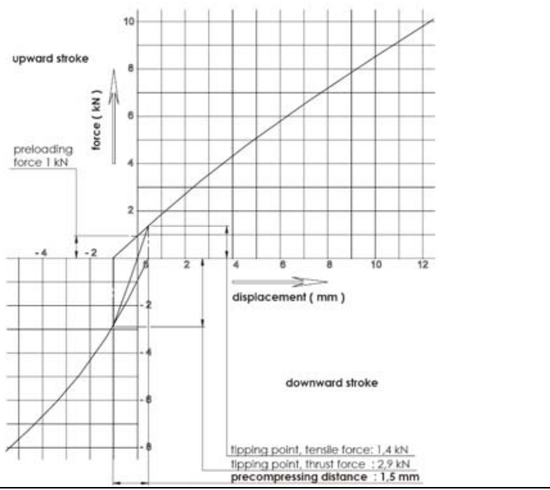
Function of shock absorber is to smoothen peak forces in pumprod. As explained in chapter 11.2, at low- and medium pumping heads, peak forces are high and elongation of pumprod is low. For those cases, a shock absorber, which provides more elongation seems useful.

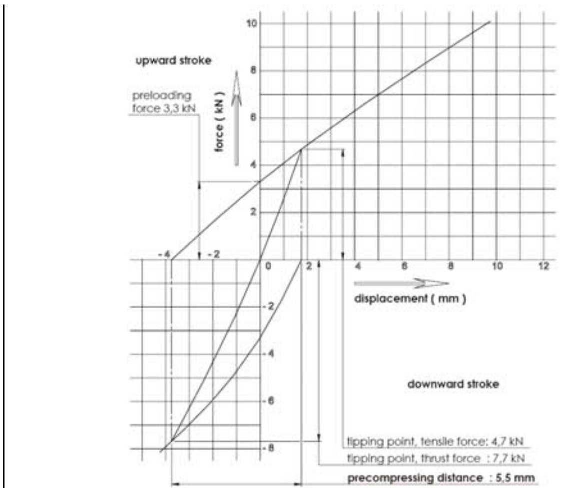


cross-section test with chain hoist installed (photo 2018)

Fig. 11.11: shock absorber

Shock absorber is fitted between pumprods, above rising main. It is equipped with disc springs which are precompressed with a certain distance. Standard fittings (2” nipple and 2” caps) are used for housing. A sliding bar, between the two pumprod connections guide the smaller disc springs and provides rigidity of shock absorber. At low wind speeds, during upward stroke, pumprod peakforces are small and shock absorber will behave stiff. Above tipping point, shock absorber lengthens more easy. During downward stroke same applies. Both tipping points are influenced by precompressing distance. Maximum elongation of shock absorber (which will occur just before breakpin of load limiter snaps) is influenced by precompressing distance.





precompressing distance 1,5mm ⇒ maximum elongation ≈ 9,5mm (when pumprodforce ≈ 8,170kN)	precompressing distance 5,5mm ⇒ maximum elongation ≈ 6,8mm (when pumprodforce ≈ 8,170kN)

Fig. 11.12: examples of precompressing distances, forces and elongations

Both pumprodconnections of shock absorber are able to rotate independently of each other. So, as a bonus, pumprodconnections cannot be unscrewed by yawing of Diever 450. Shock absorber (with precompressing distance of 3,5mm) was installed on Diever 450 on testfield in 2016. During a severe storm on 18 januari 2018 (windforce 9 with windgusts of 33m/s at testfield), pump didn’t get switched off, perhaps due to smoothened peakforce in pumprod by shock absorber. Further investigation seems useful. Weight: approx. 1,0kg.

12 Recommendations

Diever 450 is a so-called windpump; kinetic energy in wind drives a reciprocating pump, without any other energy conversions (like electricity), or gearbox in between. Pump is used to draw water from a well or a borehole, which can be used by farmers, schools or hospitals [lit. 12]. It is important, to do a feasibility study in which other pumpsystems (like: diesel- or solar pumps, small wind turbine with an electrical pump), are compared with the Diever 450, before decision is made, to use a Diever 450 for waterpumping, in case water is needed.

Diever 450 is designed and its fabrication drawings are made by students of University of Twente, Saxion, ROC and self-employed WOT members, with utmost care. Employees of aforementioned institutes kept track during the process. Prototype of Diever 450 on the testfield of WOT (which has, unfortunately, a miserable wind regime because of surrounding trees, but as a positive effect, can have very nasty fluctuations in windspeeds, winddirections and turbulences during storm conditions), has survived for more than 30 years. Without major problems. A feed-back report from Bolivia, where approximately 10 Diever's were installed, was positive about performance of Diever 450. However, WOT cannot not be held responsible for content or utilisation of any Diever 450 related literature or fabrication drawings. It is solely up to the user to "run the risk". Go for it or don't!

Chapters 8.2 and 8.3 of this report are being updated. Lit. 15 soon will provide more information concerning pumpsize and predicted output.

This report proposed a couple of changes to improve reliability and output of Diever 450, leading to different versions. For clarity, versions have been numbered according:
DA: galvanized (not yet issued)
DB: painted, as on WOT testfield
DC: as DB, with rotor improvements
DD: as DC, with possibility of balancing weight of pumprods

During writing of this report, Diever 450 (according to DD) is being built in South Africa. Results of fabrication are promising. Parts of tower are already erected. A feed-back report, about fabrication and performance (lit. 16), will be issued in due time. It is hoped, this feed-back report will enlighten WOT of effectiveness of proposed improvements, so to be able to limit versions to two max.

WOT likes to know what is going on. Members of WOT are all volunteers participating a non-profit organisation (one even has to pay a yearly sum to be accounted for being a member of WOT!). Nothing is harvested. There is no gain. Except for feed-back from you, about your experiences concerning Diever 450, in that way enabling WOT to improve services. So please do!

13 Literature (Diever 450 related)

1	Beveiligingsmechanisme voor een waterpompende windmolen *Design hysteresis safety system, head, drawings*	Chris Vos	WOT 1987
2	Logboek Diever 450 *Journal of prototype on testfield*	various authors	WOT 1987- now
3	Fabrication drawings Diever 450 *With inclined hinged main vane safety system*	Gerard Wijbenga	WOT 1990
4	Technical Report 1990 Diever 450 *Helicopter view*	Frans Brughuis	WOT 1990
5	Diever 450 herontwerp *Design hinged side vane safety system, transmission and head*	Frans Brughuis	WOT 1991
6	De nieuwe kop van de Diever 450 *Strength calculations head for hinged side vane safety system*	Arwin Baauw	WOT 1992
7	Informe sobre la construccion de aerobombas Diever 450 *Feedback-report*	CASAM	Bolivia 1992
8	The tower of the Diever 450 windmill *Strength calculations*	Frans Brughuis	Holland 1994
9	Aanpassing van de Diever 450 *Design transmission, strength calculations, drawings*	Chris Vos	WOT 2004
10	De aangepaste “Diever 450” *Design rotor, drawings*	Koen Journée	WOT 2004
11	Toren en staart “Diever 450” *Drawings tower and tail*	Richard Zander	WOT 2005
12	Van wind naar water *Comparing survey to verify the applicability of Diever 450 regarding technical, producible and commercial aspects.*	Dineke Voogt	WOT 2005
13	Fabrication drawings Diever 450 *With hinged side vane safety system versions:DA, DB, DC, DD*	Chris Vos	Holland 2017
14	Technical Report 2018 Diever 450 *Chopper view*	Chris Vos	Holland 2018
15	Pumpsize and output Diever 450 *Calculations and predictions*	Sem Riemens	Holland forthcoming
16	Experiences with Diever 450 *Feedback-report* *fabrication and performance*	Bennie Oberholster	South Africa forthcoming