

Large bearings with incorporated gears, high stiffness and precision for the Swedish Solar Telescope (SST) on La Palma

SPIE vol. 6273, *Optomechanical Technologies for Astronomy*, paper 119, 2006

Robert H. Hammerschlag^{*a}, Felix C.M. Bettonvil^{a,b}, Aswin P.L.Jägers^c, Göran B. Scharmer^d

^aAstronomical Institute, Utrecht University, Princetonplein 5, 3584CC Utrecht, the Netherlands

^bNetherlands Foundation for Research in Astronomy (ASTRON), P.O.Box 2,
7990AA Dwingeloo, the Netherlands

^cInstrumental Group Fysica (IGF), Utrecht University, Sorbonnelaan 4,
3584CA Utrecht, the Netherlands

^dThe Institute for Solar Physics of the Royal Swedish Academy,
Alba Nova, SE-10691 Stockholm, Sweden

ABSTRACT

The 1-meter Swedish Solar Telescope (SST) obtains images of the solar surface with an unprecedented resolution of 0.1 arcsec. It consists of a relatively slender tower with on top only the vacuum turret for reflecting downward the solar beam and no protective dome. This is a favourable situation to get good local seeing. Just in the case of some wind, seeing is best for daytime observations, therefore the precision bearings and drives of the elevation- and azimuth axis of the turret have to be stiff against wind. This requires line contact between the meshing teeth of the large gear wheel and the pinion. High preload forces to achieve line contact are not allowed because of appearing stick-slip effects. To reduce the risk on stick-slip a special design of the teeth for high stiffness combined with low friction and smooth transition from one tooth to the next was made. Furthermore, extreme precision in the fabrication was pursued such that relatively small contact forces give already line contact. This required a special order of the successive fabrication steps of the combination of bearing and gear teeth. An additional problem was the relatively thin section of the bearings required for a compact turret construction, needed for best local seeing and minimum wind load. Solutions for all these problems will be discussed. For the large gears the exceptional good DIN quality class 4 for the pitch precision and straightness plus direction of the teeth faces was achieved.

Keywords: telescope drives, telescope bearings, telescope stiffness, stick-slip motion, solar telescope, high resolution, seeing, telescope optics

1. INTRODUCTION

The SST¹ is located at the Observatorio del Roque de los Muchachos (ORM) on the north-western Canary island La Palma, in latitude 28.75° N, at an altitude of 2360m. Excellent seeing conditions are possible when the air mass comes from the northerly directions, between west and north-east, e.g. the directions from where the wind comes directly over the ocean, see Fig. 1. Wind speeds of just a few m/sec (e.g. 2 m/sec) up till 20 m/sec allow these excellent conditions.

The height of the temperature disturbed layer, due to forced convection near the ground, is not large. The height of 17 m of the SST tower is sufficient to be above this layer at this favourable location. This limited height permits a stable tower with the limited cross-section of 5×5 m. Consequently, the surface of the tower which causes an upward wind flow, is small and so is the temperature disturbance of the air around a telescope on top of this tower.

The foregoing has been proven by results with a previous telescope on this tower, the SVST, a 50 cm instrument with a free aperture of 47 cm. The SVST was a vacuum telescope with a doublet lens as entrance window to the vacuum and

^{*} R.H.Hammerschlag@astro.uu.nl; phone and fax +31 30 2535220; <http://dot.astro.uu.nl>

two flat folding mirrors in the vacuum which directed the light downward into the tower. The solar image was formed in the basement of the tower by the lens on top. Near this image the exit window of the vacuum was situated, where also an optical lab with light splitting, spectral analyzing and re-imaging set-ups were placed.



Fig. 1. The Swedish Solar Telescope (SST) at the Observatorio del Roque de los Muchachos (ORM) at an altitude of 2360 m. The area is free to the ocean between west (left) and north-east. Wind speeds of just a few m/sec (e.g. 2 m/sec) up till 20 m/sec from over the ocean or from over a layer of clouds above the ocean permit excellent seeing conditions. The relatively slender tower of 17 m height (right) has on top the so-called Turret, which consists of an entrance lens of 108 cm – clear aperture 97 cm – and two folding mirrors under 45° of 1.4 m diameter. This system directs the light downward through a steel tube in the tower to an optical lab in the basement of the building. The light path from the entrance lens to the optical lab is evacuated to eliminate image disturbance from air-temperature fluctuations inside the telescope. The entrance window is visible as a black ellipse on top. The Turret is made as compact as possible around the optical beam. Consequently, its outside shape shows the two folding mirrors under 45° with one pivot-joint with the horizontal axis – the elevation axis – between the two mirrors and another pivot-joint with the vertical axis – the azimuth axis – just underneath the second mirror. The flat round piece to the left is a counterweight to balance the azimuth axis. The lid to cover the entrance lens can be seen with its support system on top of the tower at the very right. At the left side of the picture one can just see the Dutch Open Telescope (DOT), likewise a high-resolution solar telescope, but with a contrasting principle: the primary light beam 'inside' the telescope is as open as possible for air flushing by natural wind. Both principles prove to be very successful on La Palma. Despite the telescopes' difference in principle, the SST gears are a further development of these of the DOT.

The SST is an enlargement to a 1 m instrument with a free aperture of 97 cm and the proven concept of the SVST. A schematic set-up of the SST is shown in Fig. 2. An important difference with the SVST is the use of a singlet lens instead of a doublet, due to production problems and the limited wavelength range of diffraction limited quality for a doublet of 1 m diameter.

The chromatic aberrations of the singlet are corrected by an additional optical system, the Schupmann corrector. A field mirror reflects the light back in the primary image – see detail A in Fig. 2 – towards a correction system of a negative lens and mirror, which is used in nearly auto-collimation – see detail B in Fig. 2. The primary image is re-

imaged with a magnification of one to one next to it, where the light goes through the exit window into the optical lab. There, the light first is reflected by a tip-tilt mirror to correct the image motion and then by a deformable mirror for wavefront correction. Next the light goes through a lens for re-imaging – see detail C in Fig. 2. The tip-tilt and deformable mirror are part of an adaptive optics (AO) system.²

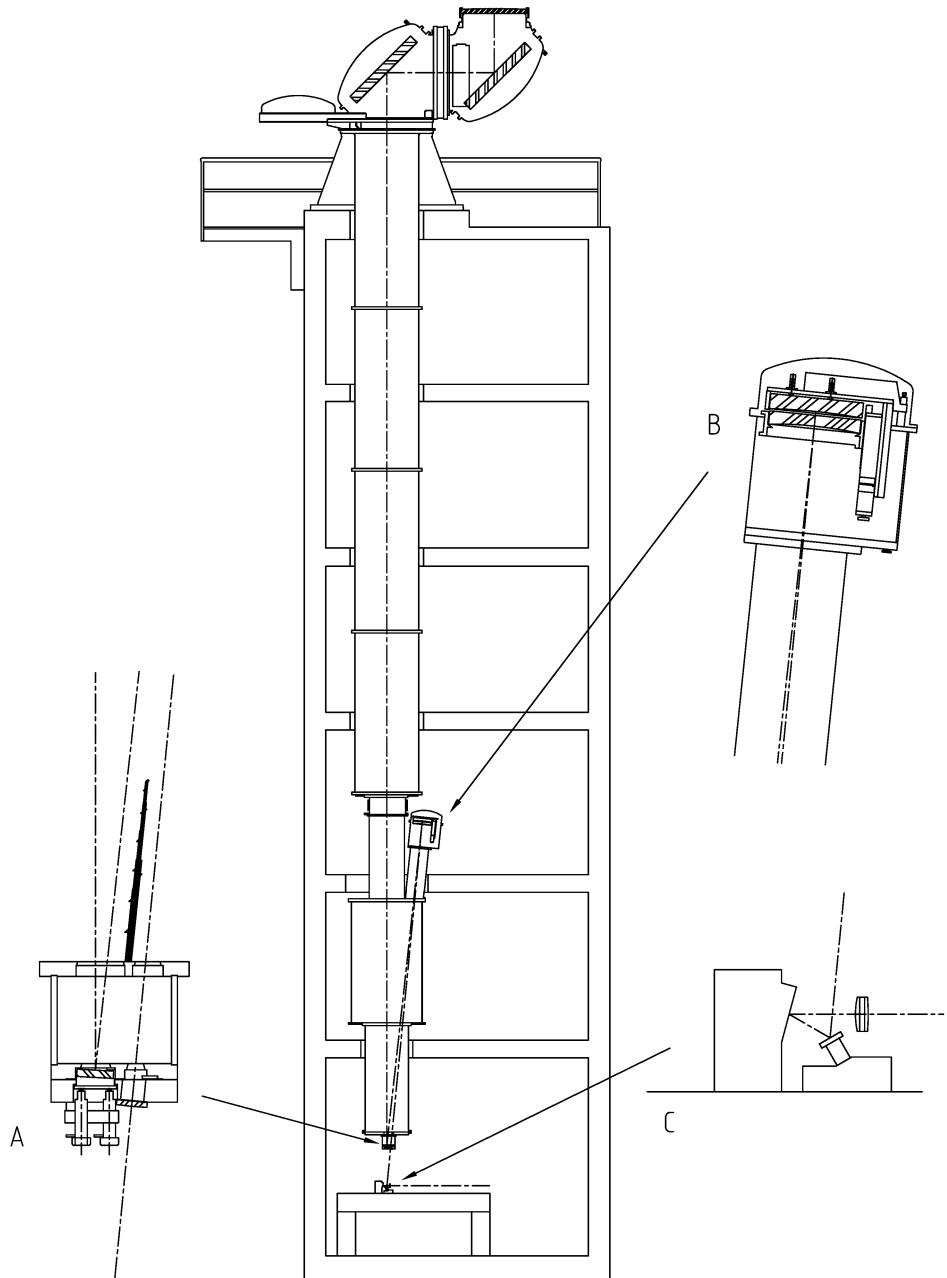


Fig. 2. Schematic set-up of the SST. The successive optical and motion components are: singlet lens as entrance vacuum window and optical pupil – first folding flat mirror – pivot-joint with horizontal axis (elevation) – second folding flat mirror – pivot-joint with vertical axis (azimuth) – in detail A: primary image on the field mirror at the left side – in detail B: pupil image with Schupmann corrector consisting of a negative lens with focusing stage, followed by a concave mirror in nearly auto-collimation – in detail A to the right: the corrected 1:1 image on the field lens as exit vacuum window – in detail C: second pupil image with tip-tilt mirror followed by the AO deformable mirror and re-imaging lens.

2. THE TURRET DESIGN

The light path of the system is shown in Fig. 2. On top is the singlet lens, which is at the same time the entrance window for the vacuum. The fused silica lens has a diameter of 108 cm and a clear aperture of 97 cm. The light goes from the lens to a first flat zerodur mirror of 140 cm diameter, tilted to 45°. This first mirror reflects the light in a horizontal direction towards a second identical mirror, which reflects the light downward into the tube to the optical lab.

Between the first and second mirror a first pivot-joint with large bearing is placed with a horizontal axis, the elevation bearing, because a rotation of the lens plus first mirror points the telescope beam in elevation. Under the second mirror a second pivot-joint with bearing is placed with a vertical axis, the azimuth bearing. Both mirrors and the lens rotate around this vertical axis. The light beam of about 1 m diameter passes in vacuum through these two large ring-shaped bearings.

Consequently, the pointing Turret on top of the tower consists in principle of 3 parts:

- Top part or Elevation mirror house with lens and first mirror,
- Mid part or Azimuth mirror house with second mirror,
- Under part or Stativ, which forms the connection to the tower.

A composition drawing of the Turret with the dimensions is shown in Fig. 3, whereas Fig. 4a,b shows photographs of the Turret during tests in the machine-shop of Svenska Bearing AB, where the Turret was assembled.

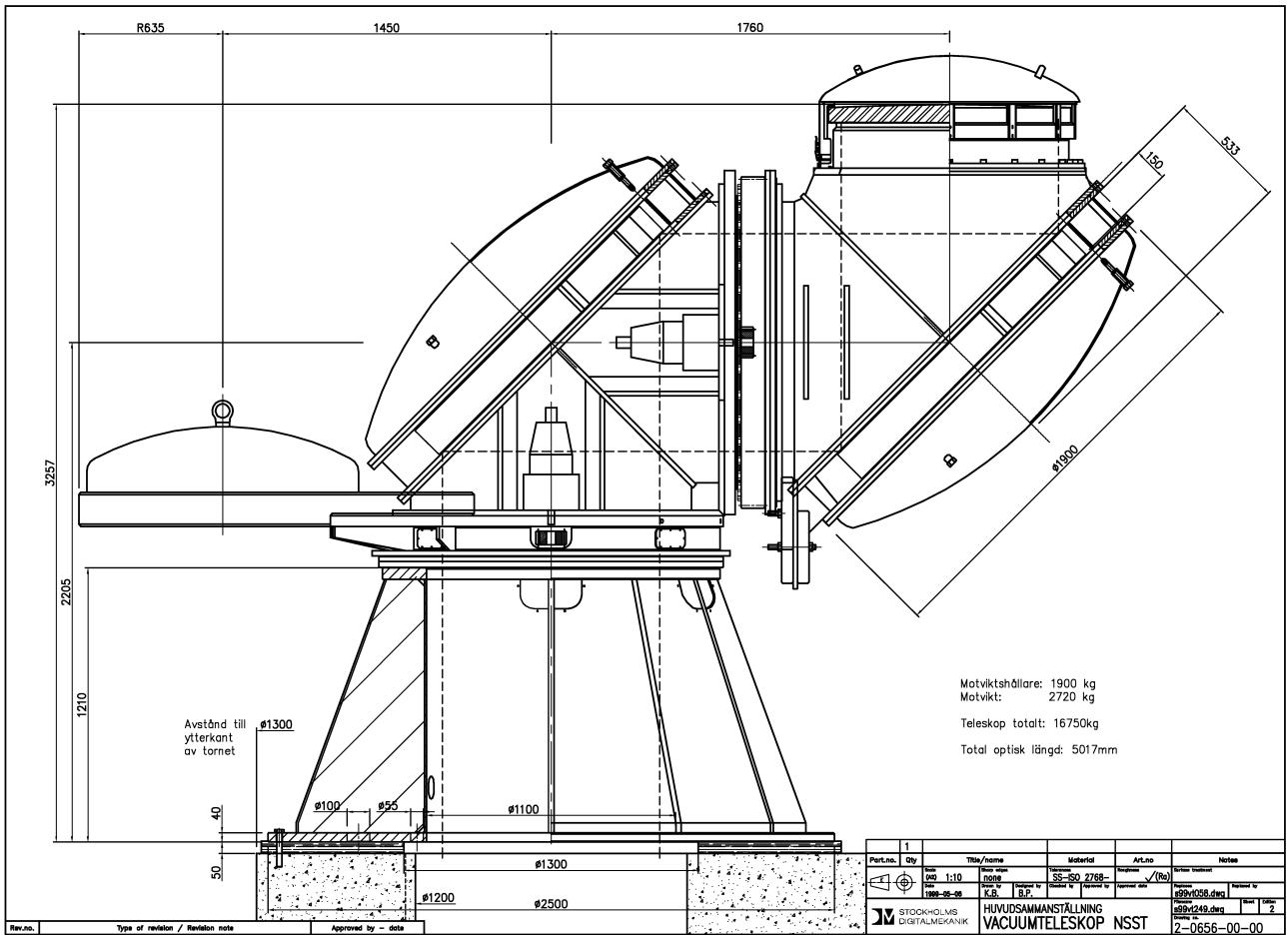


Fig. 3. Composition drawing of the Turret: *to the right* Top part with lens and first mirror – *in the center* Mid part with second mirror, drive motors for both elevation and azimuth – *to the left* the counterweight which belongs to the Mid part for balancing the azimuth axis – *at the bottom* the Under part which forms the connection to the tower.

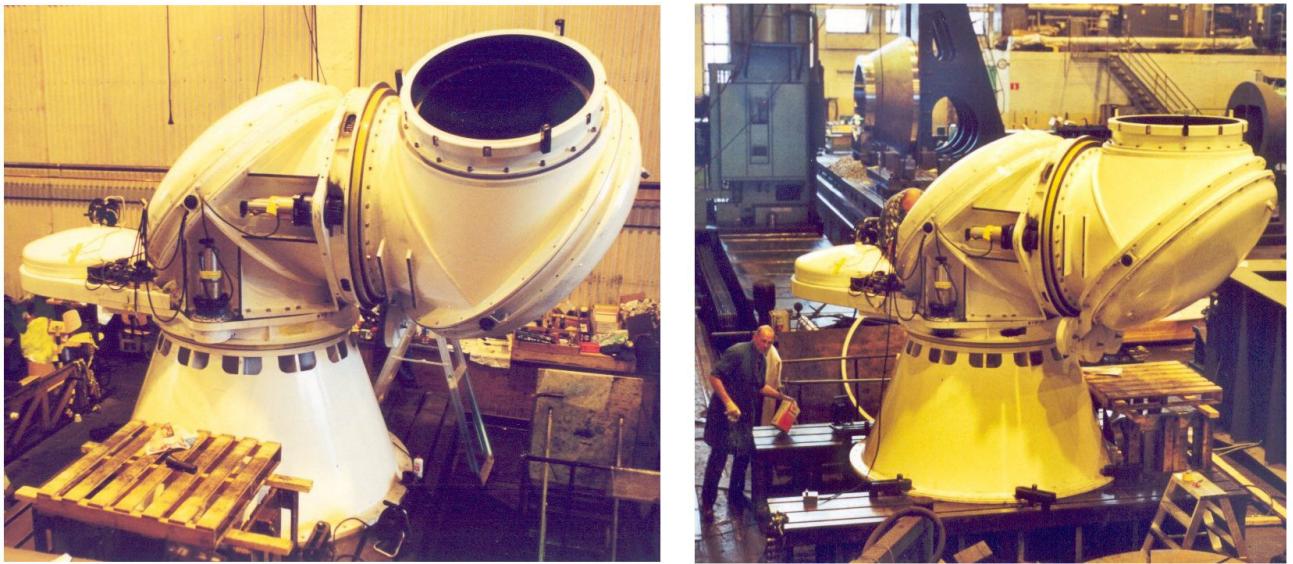


Fig. 4. Turret during tests in the machine-shop of Svenska Bearing AB, where it was assembled. (a) Front view with open pinion of the elevation drive; (b) view with both pinions open; to the left Piet Hoogendoorn greasing parts, half hidden on the counterweight the electronics- and software engineer Peter Dettori and at the background one of the huge milling and cutter machines of Svenska Bearing AB, where all the large parts for the Turret were made.

The goal of the design was to make the construction as compact as possible in order to reduce the wind load. In addition, such a design is favourable to obtain a minimum disturbance of the airflow. Temperature fluctuations of 0.1°C of the air itself in the incoming optical beam of 1 m diameter are already harmful for the image quality. The structure parts show easily temperature differences of a few degrees Celsius because of their different angle to the solar radiation (heating) and to the incoming-wind airflow (cooling). The latter can also show changes in temperature: tenths of a degree Celsius over timescales of minutes. The small dimensions of the Turret have the effect that, with a wind of only a few m/sec, the incoming air does not have sufficient time to be heated in a harmful way. This compact construction gains an advantage over telescopes in a dome and in this respect the SST is an 'open telescope'.

The previously mentioned goal left us with the task to built a compact construction which is still very precise and stiff around the light beam. The two large bearings between Top part, Mid part and Under part inevitably have to become thin rings, which are not stiff from themselves. Chosen are roller bearings with many small cylindrical rollers between very precise raceways, see Fig. 5. In this application, roller bearings are preferable to hydrostatic bearings because of the heat production of the hydrostatic pump, which would be harmful to the air temperature around the telescope. Moreover, there is not much space for a drive and in the chosen solution the gear teeth are placed directly on the bearing rings.

In such a design, the slender bearing rings have to get their stiffness from the surrounding construction, in this case the 3 mentioned parts. Their stiff constructions, which still are compact, are visible in Fig. 6 for the Top part, in Fig. 7 for the Mid part and in Fig. 8 for the Under part. A detailed Finite Element Analysis was made by High Tech Engineering in Stockholm, see section 3.1 of ref¹.

The bearings are placed into separate ring-shaped houses, see Fig. 9 with sub-composition drawings of the bearing construction, of which a detail-cut is shown in Fig. 5. The separate bearing houses enable an easy assembling on the observatory mountain at La Palma. There were only 5 big parts, viz. the Top-, Mid- and Under part and the 2 bearing houses, all 5 of moderate size and weight. These could be handled by a crane which was able to stand at the difficult reachable location of the telescope. Notice the many screws needed for the connection of the 5 parts in order to transfer the stiffness of the Top-, Mid- and Under part to the 2 bearing houses over the whole circumference.

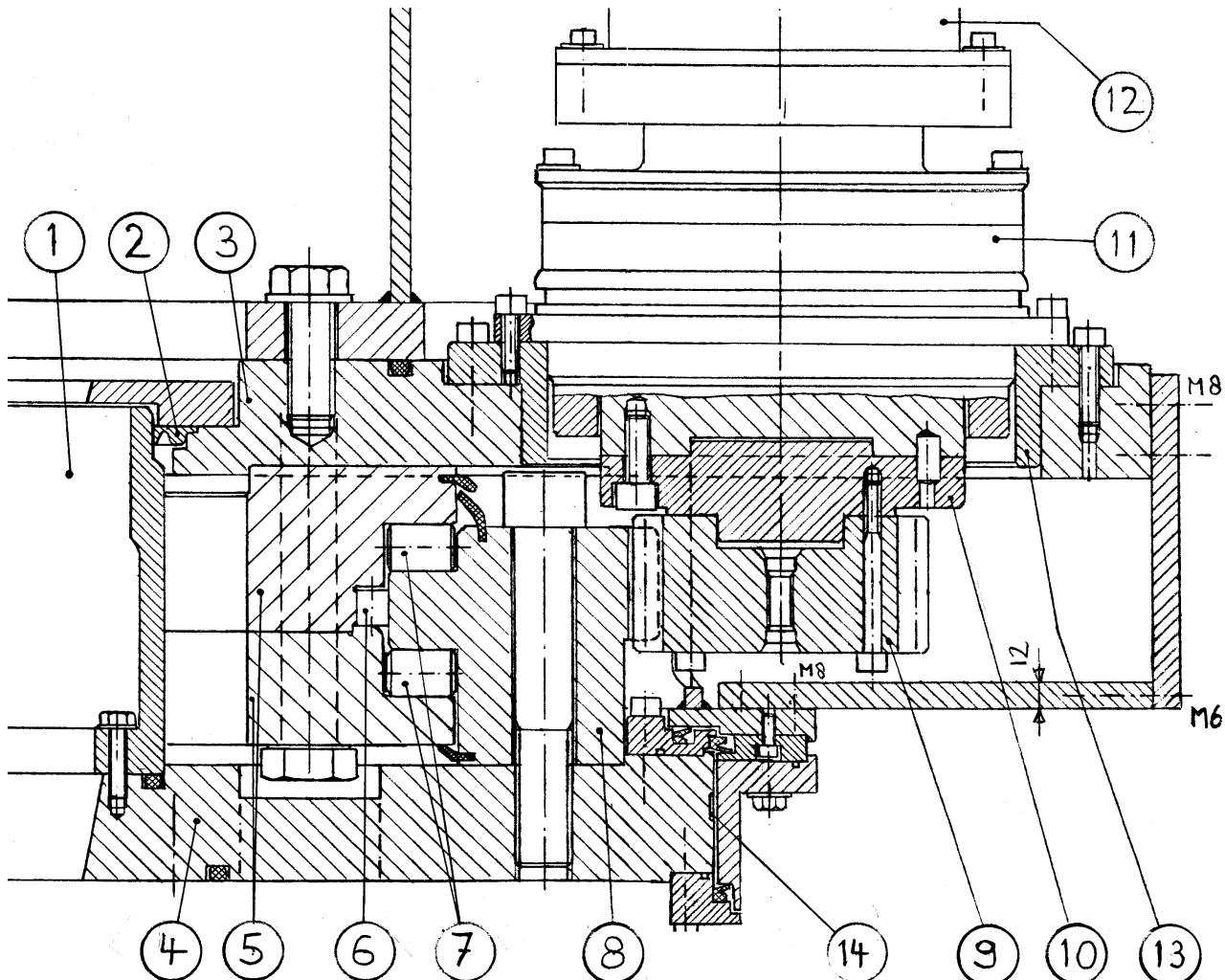


Fig. 5. Bearing design with incorporated gears for the elevation- and azimuth pivot-joint around the light beam in vacuum.

- 1 Vacuum region with primary light beam.
- 2 Vacuum seal.
- 3 Bearing house part, which is connected to Mid part of Turret. This part includes a welded ring house around the bearing.
- 4 Bearing house part, which is connected to Top part of Turret for the elevation, or, to the Under part of the Turret for the azimuth. This part consists only of a thick plate.
- 5 Split inner ring of bearing, via bearing house part connected to Mid part of Turret.
- 6 Row of radial rollers.
- 7 Two rows of axial rollers, larger in diameter because the axial forces, of which a substantial part is caused by moments, are larger than the radial forces.
- 8 Outer ring of bearing, via bearing house part connected to Top part (elevation) or Under part (azimuth) of Turret. Gear teeth are at the outside rim, module $m=6$, number of teeth $Z_2=238$, tooth width $b_2=50$ mm, outside profile shift $x_2=1.15$.
- 9 Pinion, number of teeth $Z_1=19$, tooth width $b_1=60$ mm, outside profile shift $x_1=0.363$.
- 10 Connection piece between pinion and outgoing shaft of planetary gear.
- 11 Planetary gear drive, two stages, reduction ratio $i=31$, type TP110-MF2-31 from Alpha Getriebbau; a low-backlash compact drive combining high stiffness with low friction.
- 12 Servomotor with added 3-stage drive of spur gears, reduction ratio $i=20$.
- 13 Connection ring between house of planetary gear drive and the bearing house part, which is connected to the Mid part of Turret.
- 14 Encoder steel scale tape of Heidenhain.

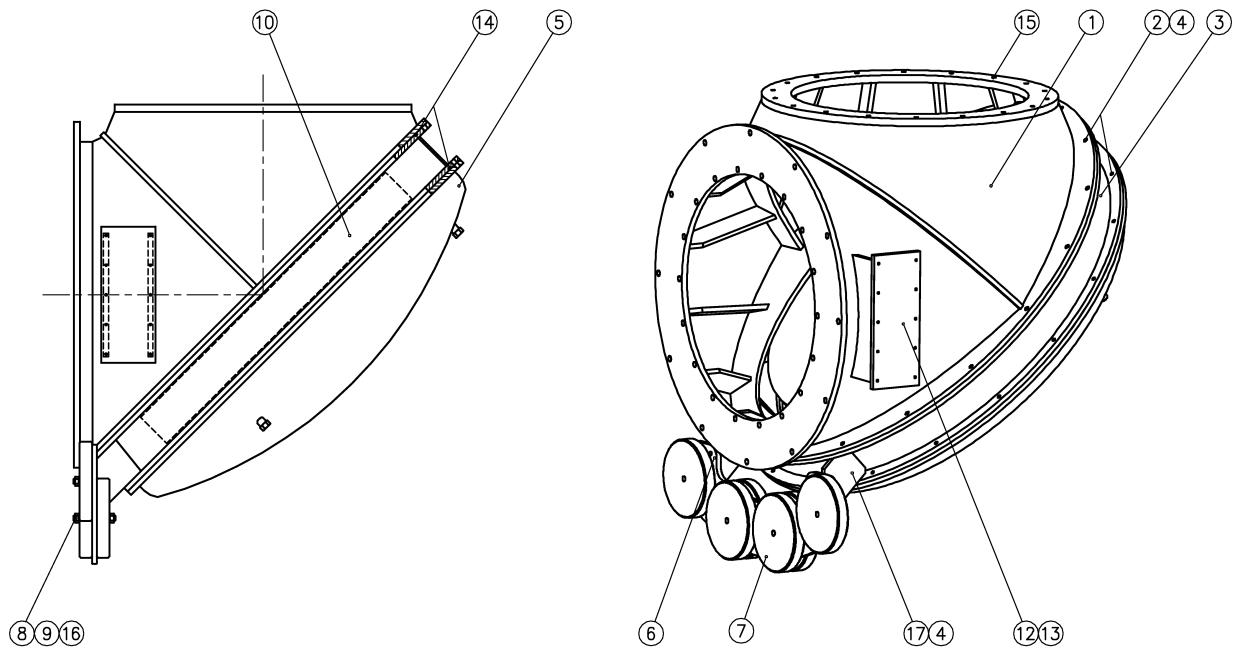


Fig. 6. Top part of the Turret. The mount of the lens connects on top of this part, the elevation bearing house to its left side. The first flat folding mirror, i.e. the elevation mirror, is located under 45° upward at the right side of this part. Notice the stiffening plates inside the house, close around the light beam. This compact, but still stiff construction combines a low area exposed to the wind with a stiff support of the bearing house. The 4 round plates at the bottom are counterweights to balance the elevation axis.

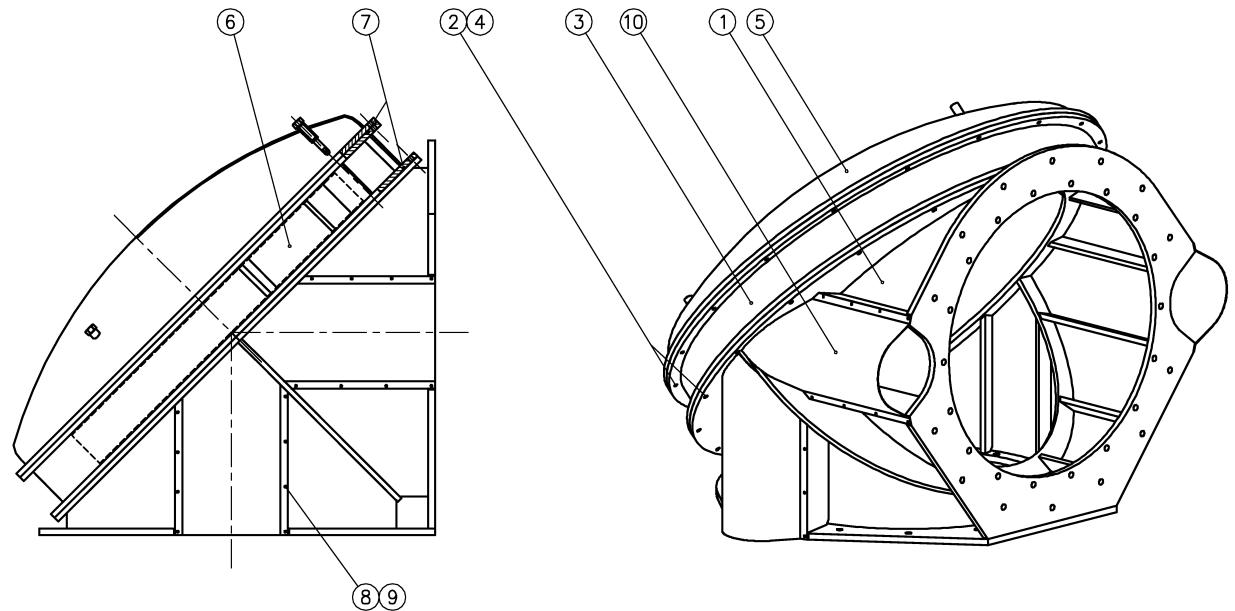


Fig. 7. Mid part of the Turret. The elevation bearing house connects to the right side of this part, the azimuth bearing house to its bottom side. The second flat folding mirror, i.e. the azimuth mirror, is located under 45° downward at the left side of this part. Notice the stiffening plates inside the house; for an explanation see Fig. 6.

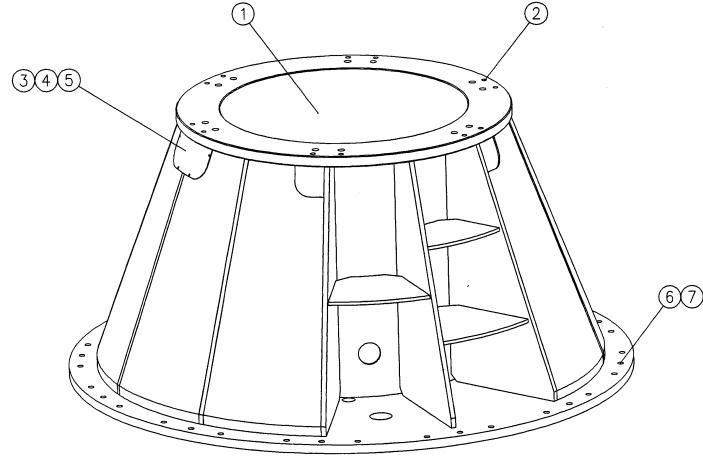


Fig. 8. Under part of the Turret. The azimuth bearing house connects to the top, the tower roof to the bottom side of this part. Notice also here the stiffening plates inside the construction (cf. Fig. 6 and 7).

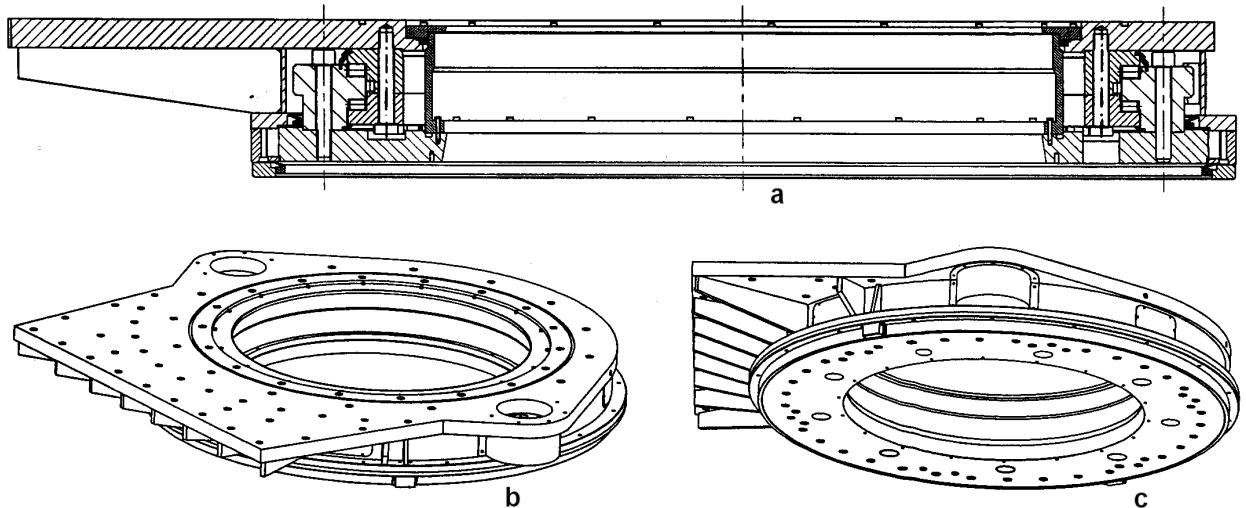


Fig. 9. Sub-composition drawing of the azimuth bearing house: (a) cross-section; (b) 3D-top view; (c) 3D-bottom view. The cantilever at the left serves for the connection of the counterweight to balance the azimuth axis. The construction of the elevation bearing house is similar, but without cantilever for counterweight. The house is circle-symmetric with only two broadenings for the two pinions. The bearing houses give radial stiffness to the bearing rings. The axial stiffness and some additional radial stiffness come from the Top-, Mid- and Under part.

3. THE BEARING DESIGN WITH INCORPORATED GEARS

A bearing consists of two circles of larger sized rollers for the axial loads and one circle of smaller sized rollers for the radial loads, see Fig. 5. The inner ring is split into two halves, which are screwed together and which lock up the outer ring between the circles of axial rollers. All parts are grinded very precisely in such a way that all the rollers, both the radial and axial ones, are under a preload somewhat higher than the largest forces which can be expected.

The rings are rolled in their shape whilst hot and are pre-turned. The material used is 42CrMo4 steel which is quenched and annealed to a hardness of 287 HB, corresponding to a strength of 970 N/mm². The raceways for the rollers are induction-hardened. Only a surface layer becomes very hard like the rollers, the rest of the ring remains tough.

The compact design required very precise gear teeth directly in the outer bearing ring. The teeth are not hardened, but have the hardness of the quenched and annealed base material. These teeth mesh with two pinions on opposite sides, hence 180° apart. The pinions are made of the same material as that of the rings. After fabrication of the rings, their hardness was measured and the pinions were quenched and annealed to exactly the same hardness. The gears in the telescope run so slowly, that no real lubrication film of oil is being built up. Wear is minimized when the large wheel and pinion both are of the same steel type and of equal hardness. A hardened pinion would cause increased wear on the large gear. This is different from the case of fast-running gears, where a hardened pinion is an advantage. The foregoing was demonstrated by investigations at the Gear Research Institute of the Technical University München. To obtain low wear it would be best to use hardened teeth for both pinion and large gear wheel. However, hardened teeth for the large gear wheel are very expensive, because in that case a much larger grinding depth is needed due to the deformations caused by the hardening of a large, relatively slender piece. The forces in the telescope are moderate and therefore hardened teeth are not required.

Grinding the teeth in the outer ring before the assembly of the bearing, would give the risk of teeth that are not exactly aligned with the bearing because of the slender, compact construction of the rings, which get their final stiffness from the Turret parts. The alignment requirement is very high, because to obtain a large stiffness the meshing teeth of large wheel and pinion have to make contact over their full width for all positions of the large wheel. The meshing teeth have to be parallel within a few micrometers, see next section.

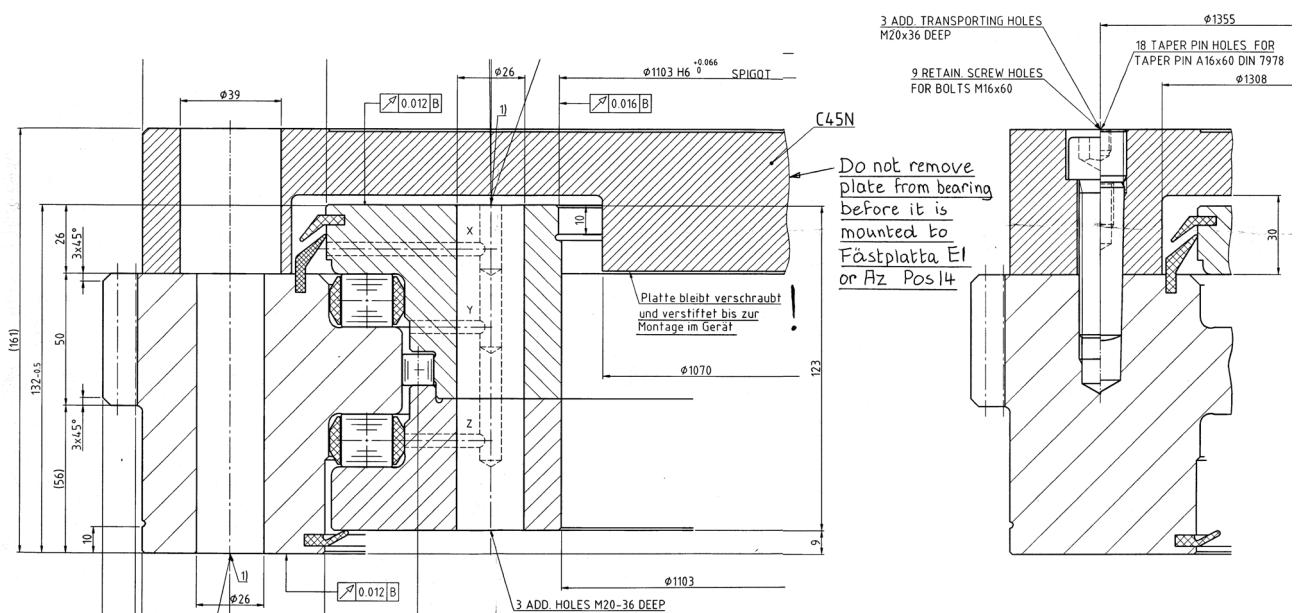


Fig. 10. Completely assembled bearing with on top an auxiliary thick plate, connected to the outer bearing ring with 18 taper pins Ø 16×60 mm and 9 bolts M16×60, see cross-section at the right. This plate procured radial stiffness to the bearing during the whole further process. For the finish-grinding of the teeth, the whole unit as shown in this figure was clamped with the bottom side of the outer bearing ring on the table of the tooth grinding machine. In this way the bearing was put in a completely closed space whereto the grinding dust and cool water had no access. The bearing was assembled into the telescope by screwing the complete unit to the corresponding bearing house part with M24×150 bolts with a cylindrical head and an inside hexagon for a key (ISO 4762, DIN 912). The heads of these bolts went through the holes Ø 39 mm in the thick plate, see cross-section at the left side of the left top corner. After tightening these bolts, the plates were dismounted by taking out the taper pins and M16×60 bolts.

The chosen solution was the following:

1. The gear manufacturer pre-toothed the outer bearing rings by milling.
- 2a. The bearing manufacturer grinded the outer rings ready. In this way small deformations in the rings, caused by the tension from the gear milling, were corrected.

- 2b. The bearing manufacturer assembled the complete bearing on an auxiliary thick plate screwed on top of the outer ring, see Fig. 10. This plate had two functions: (1) it procured complete radial stiffness to the bearing during the whole further process, including the final assembly in the bearing house of the telescope and (2) it protected the bearing against dirt and water during the final grinding of the teeth. Test measurements were made for the completed bearing on the thick plate.
3. The gear manufacturer finish-toothed on the complete bearings by grinding. Here, only a small grinding depth was required, see our earlier remarks about the not-hardened teeth.
 4. The bearing manufacturer repeated the test measurements on the bearings. It was found that the precision of the bearing did not decrease by the grinding of the teeth.
 5. Each bearing, together with its thick auxiliary plate, was mounted on the corresponding bearing house part. The auxiliary plates had additional borings for the heads of the necessary bolts, see Fig. 10. After tightening these bolts, the auxiliary plates were dismounted.

4. GEAR TEETH FOR HIGH STIFFNESS AND LOW FRICTION

The teeth are relatively short, resulting in a high bending stiffness. Consequently, the contact length along the line of action of meshing teeth reduces. This is not a disadvantage in the case of the low telescope velocities as long as the contact length is clearly longer than one pitch.

Gear teeth inevitably have small pitch deviations when going from one tooth to the next. A displacement in the telescope pointing of one tenth of an arcsec corresponds to the μm -region measured along the circumference of the large gear wheel, i.e. the same region as the remaining pitch deviations. These deviations can cause motion irregularities in the transitions from one pair of meshing teeth to the next one.

A small tip relief of the large wheel- and pinion teeth brings about smooth transitions. The relief is much smaller and less deep than the relief for compensation of the tooth bending in fast running turbine gears. For the telescope, the necessary relief is so small that it does not have to go deep along the tooth surface. The relief is between 0.025 and 0.050, nominal 0.037 mm, with a length along the contact line of 2.16 mm nominal. The corresponding depth along the radius becomes 1.20 mm for the pinion and 0.83 mm for the large wheel. Tooth height and relief depth is chosen such that the remaining involute contact length is between 1.2 and 1.3 times the pitch length. This is a good compromise between short teeth for stiffness and smooth transitions.

The relief has the shape of a rounding which connects to the involute of the tooth profile without angle. The radius of the rounding is 16 mm for the pinion and 12 mm for the wheel. The negative of the tooth profile with relief is made in the grinding stone with a diamond tip on a numerical machine. Hence, it is a profile grinding and not an unroll grinding. The grinding goes to the bottom of the tooth spaces, which is better for stiffness and strength than milled free cuts at the base of the flanks. Unroll grinding also would be possible to the bottom of the tooth spaces, but it requires more labour.

The teeth of the pinion have a positive profile displacement to the outside in order to increase the curvature radius in the used region of the involute. With a low number of teeth the curvature radius is small and the Hertz contact pressure high. A positive profile displacement increases the strength and stiffness due to a smaller local elastic deformation.

In machine engineering it is normal practice to use a positive profile displacement only for the pinion, because it is not necessary for the strength of the teeth of the large wheel. As a consequence, the contact region along the line of action shifts to one side of the rolling or pitch point. Only in the pitch point a pure rolling without sliding takes place. The sliding and, with it, the friction increase by the shift of the contact region. For normal gears, which transform power from high speed to low speed, this is not important, but for the telescope gears it is. In the following we will give an explanation.

The telescope has on each large gear wheel two pinions with separate motors which are tensioned against each other to eliminate the backlash. This is absolutely necessary because of changing temperature differences between the large gear wheel and the bearing house.

One of the motors is the driver and the other is the driven one. The gears of the driver motor are used in the normal way, i.e. from high speed to low speed. However, the gears of the driven motor are used from low speed to high speed.

This type of gear use is more sensitive to fluctuations in the friction and therefore causes an unequal rotation and in the worst case stick-slip. The function of the motor, driver or driven one, reverses with the direction of the rotation. Consequently, the gears of both motors have periods in which they are working in the speed-increase mode.

The solution for this problem is the addition of a large positive profile shift to the teeth of the large gear wheel. This shifts the contact region back to its symmetric position around the rolling point without reducing the positive effect of the increase of the curvature radius of the used involute tooth surface part for the pinion.

A large positive profile shift is allowed for gear wheels with many teeth. In that case, the tooth shape does not change much. However, gears with a smaller number of teeth get pointed teeth in the case of a too large positive profile shift. Perhaps, also for this reason a large profile shift for the larger wheel is not put into practice in normal gear engineering. However, the slowest stage of a telescope drive normally has many teeth – more than 100 – because the forces are not high considering engineering terms. Also, using many teeth for the large wheel reduces the size of the pinion.

The chosen profile shifts for the telescope gears are:

$$\text{- Pinion: } x_1 \times m = 0.363 \times 6 = 2.18 \text{ mm}$$

This value is sufficient to bring the load capacity and the stiffness of the pinion teeth near the values for the teeth of the large wheel. In addition, the safety against tooth breakage increases.

$$\text{- Large wheel: } x_2 \times m = 1.15 \times 6 = 6.90 \text{ mm}$$

This value does not shift the contact region completely back to its symmetric position around the rolling point. The result: 60% of the contact region above the rolling point for the pinion tooth faces and, consequently, 40% in the foot part under the rolling point. These numbers are measured along the path of contact, not on the tooth faces. For teeth of the large wheel these numbers interchange. For fabrication reasons we did not go to larger profile shifts. The chosen combination of profile shifts is a best compromise between many factors.

General information about the choice of profile shifts (also called 'addendum modifications') can be found in the norm paper DIN 3992. In this norm paper attention is also given to the problems of gears used from low speed to high speed. For the special case of telescope gears, however, it is better to chose profile shifts for the large wheel that are larger than in the examples of the norm. When applying profile shifts, attention should be given to the shaft distance, which does not simply increase with the sum of the profile shifts, but with a somewhat smaller value, to be calculated with the involute function.

Sudden variations of friction, in the worst case stick-slip, give problems to the control system. This makes it worth to take particular care in minimizing friction irregularities, as described above. The control system for the SST, including the two motors which are tensioned against each other on each axis to eliminate the backlash, has been described in ref³.

In principle, every drive system is a mass-spring system, see Fig. 2a,b of ref⁴. In that paper an explanation is given of the principle of stick-slip. The formulae show two methods to reduce the risk on stick-slip: (1) reduce sudden changes in the friction, like in the case of the described profile shifts; (2) increase stiffness, which is already important to reduce telescope shaking by wind gusts.

Stiffness calculations of gears are based on the assumption that two meshing teeth make contact over their full length. If the teeth are not exactly aligned or the shapes are irregular, contact is only made in a small part of the teeth. The stiffness lowers due to a larger local deformation in the smaller contact region, according to the formulae of Hertz, and because of a larger bending deformation of the smaller loaded part of the teeth. The last factor is of more importance in the case of a module which is small compared to the tooth width. However, a small module is attractive to reduce the size of the pinion for a compact design and a large reduction factor in one stage.

In normal engineering cases, the line contact is, due to deformations, reached under much heavier loads than the telescope loads. However, for telescope operation small loads are preferred, because it reduces friction fluctuations and the risk of stick-slip for the slow velocities of the telescope. The reduction of the tooth stiffness, when applying slender teeth to reach line contact under a small load, is not an option for telescopes, because the reduction of stiffness and mechanical eigenfrequency is bad for the avoidance of wind shakes and also gives problems with the control system. Such a stiffness reduction, however, is well known for a totally different and special application, viz. for extremely fast moving turbine gears.

Using precisely grinded teeth, the not-exact parallelism of the teeth forms the largest contribution to the not reaching of full-line contact under a small load. This problem has been solved without reducing the stiffness with a specially designed system of self-aligning pinions for the gears of the Dutch Open Telescope (DOT).⁵ However, the system requires more space than a simple pinion and would reduce a bit the compactness of the Turret. Since the production of the DOT-gears a significant improvement of gear manufacturing has been reached by the application of numerical machines, for instance by applying profile grinding instead of unroll grinding, the latter being a more complex combination of motions and consequently more sensitive to deviations. It was decided to try to make all the teeth of the large gear wheel precisely parallel within a few μm , viz. the deformation of the teeth under the small telescope load. In the design of the connection of the drive unit a possibility has been added for the adjustment of the pinion orientation to the tooth direction of the large wheel. To this purpose the connecting ring 13 in Fig. 5 has been made 0.05 mm wedge-shaped. By turning the ring in the right position during the assembly, it is possible to adjust the teeth of the pinion with a precision of a fraction of a μm over a range of $\pm 10 \mu\text{m}$.

After fabrication, the gear teeth were measured with special attention to the angle (direction) deviation $f_{H\beta}$ of the tooth trace over the width and to the pitch deviations f_p between two neighbouring teeth. The first deviation is important for the alignment of the teeth and consequently for the stiffness, the second one to see whether the applied tip relief is large enough. Fig. 11 shows a large gear wheel and a pinion on the measuring machines of the gear manufacturer A.Friedr.Flender AG, Bocholt. Fig. 12 shows some examples of the results of the measurements.

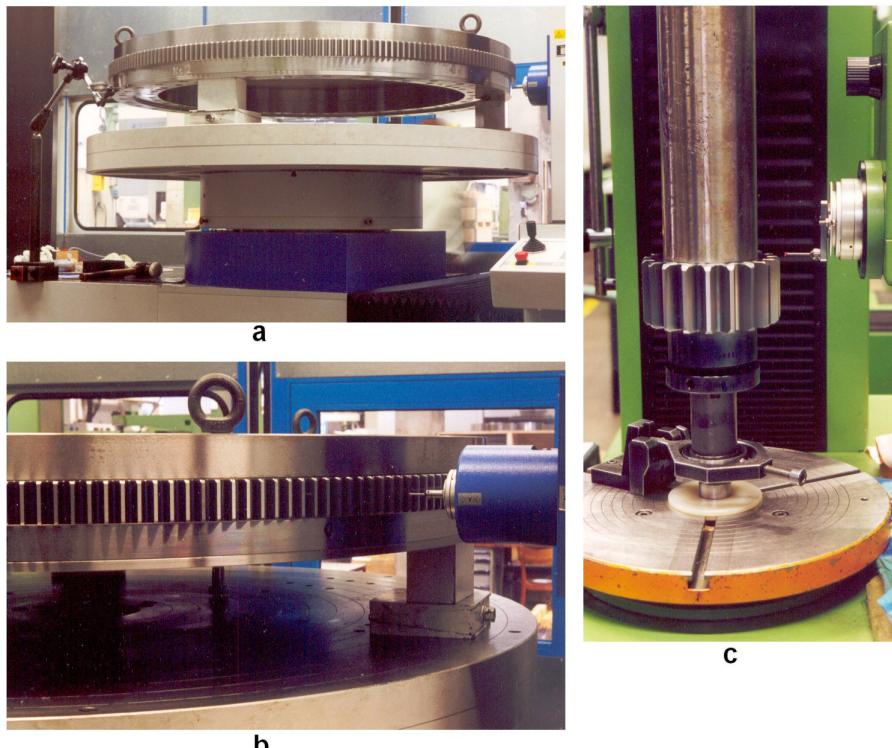


Fig. 11. Measurement of the teeth: (a) on the measuring machine one of the two large gear wheels, at the same time bearing, with thick plate on top for stiffness and protection; (b) detail with taster on tooth; (c) pinion on smaller measuring machine.

The conclusions from these measurements are:

1. Line contact over the tooth width can be expected.
2. No pinion adjustment is necessary as far as the large bearing + gear wheel and pinion are concerned. Of course, deviations in the bearing house and in parts of the drive unit could still require an alignment with the help of ring 13 in the drive unit, see Fig. 5.
3. The applied tip relief is comfortably large.
4. The gears are within class 4 of DIN 3962 for the deviations which are of interest to our application concerning the tooth trace and pitch. A lot of information about gears can be found in three DIN pocketbooks.⁶

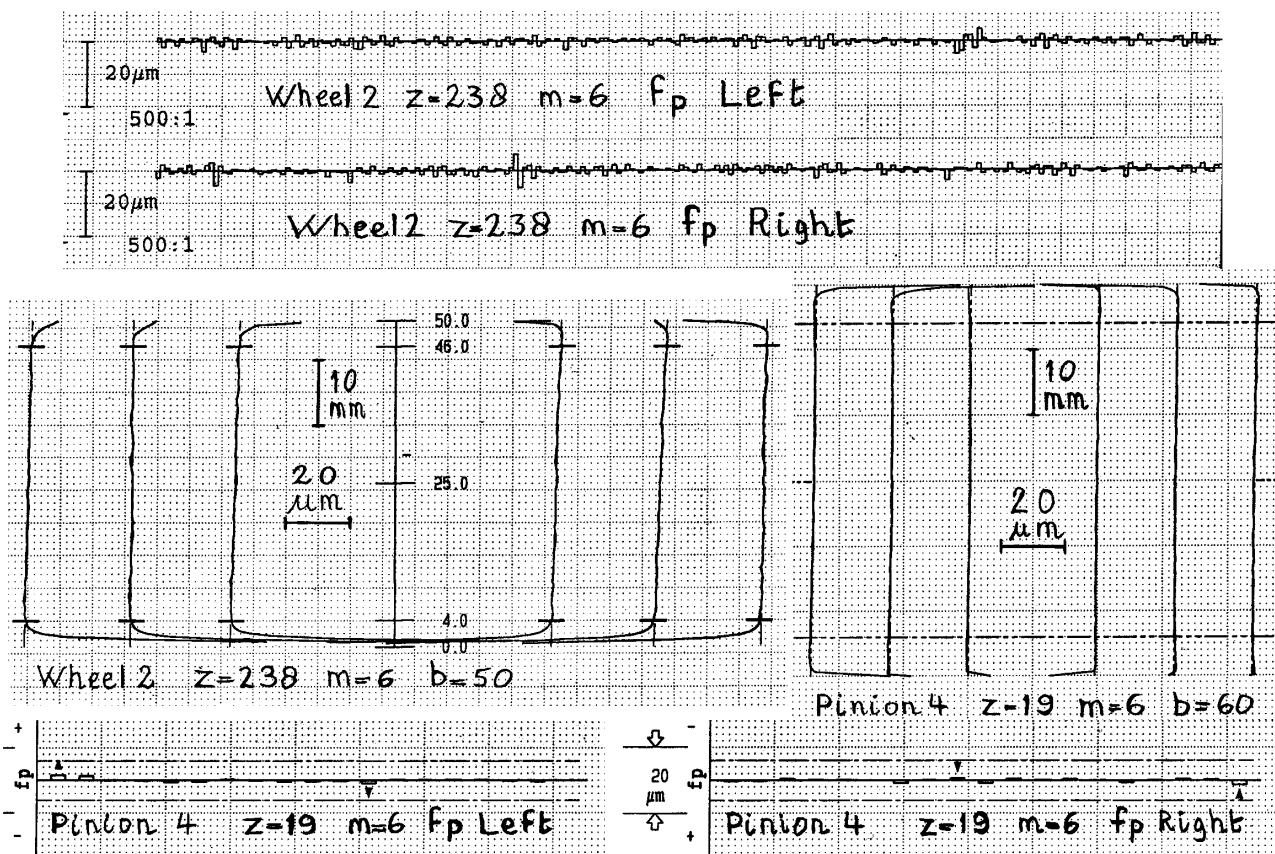


Fig. 12. Results of the measurements. Top: pitch deviations of the wheel. Center left: tooth traces over the width of the wheel.

Center right: tooth traces over the width of the pinion. Bottom: pitch deviations of the pinion.

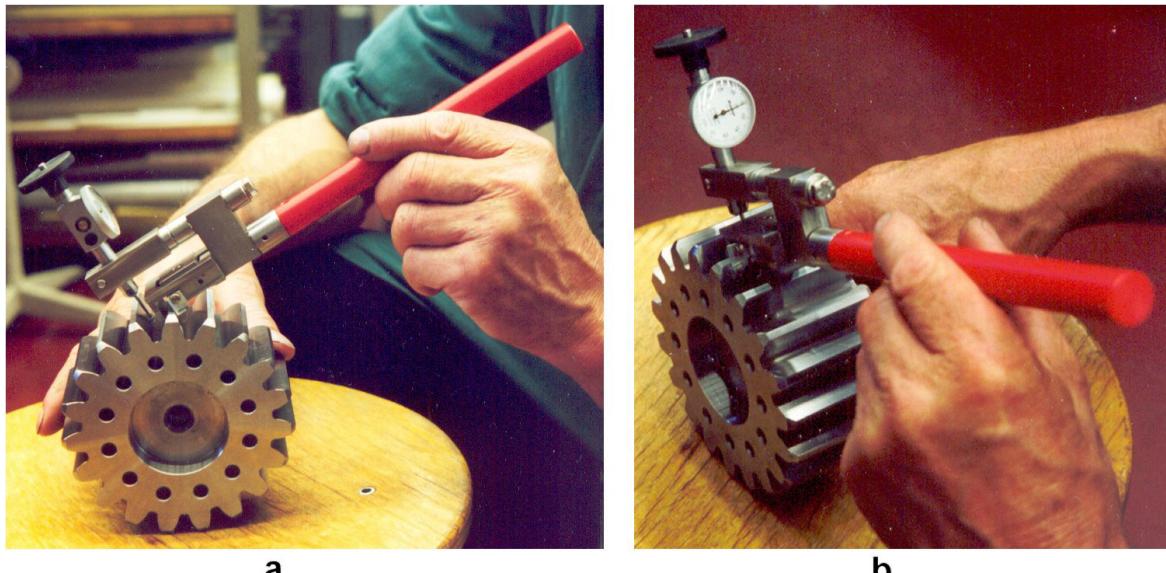


Fig. 13. Pitch gauge: instrument to measure with μm precision the pitch deviations of a pinion in a direct way and exactly as seen by the counter gear. (a) Side view showing how the instrument works. A small block is pressed against a tooth – the height is adjustable – and another small block behind the tooth. The instrument is so-to-say lying on a tooth. The handle is moved up and down and the maximum push-in of the gauge corresponds to the minimum distance, i.e. the pitch. (b) Front view with gauge.

In addition, the pitch deviations between neighbouring teeth were measured with a simple home-made, but still very precise instrument, see Fig. 13. It measures the deviations in the pitches directly and exactly such as seen by the counter gear, i.e. the contact pitch deviations f_{pe} . The theoretical contact pitch is equal to the transverse base pitch. Many years ago the Swiss gear manufacturer Maag made an instrument based on the same principle. However, Maag gears finished its activity a long time ago and, in addition, our home-made instrument is much easier to handle than the much heavier devices of Maag, see the old pocket-book of Maag⁷. As a consequence, with this instrument gears can be reached at difficult places; only access from the head-side of the gear is needed. The Maag devices had a read-out of the gauge from the side. The home-made pitch gauge provided an independent fast control with μm precision on site, without the need of complex machines with electronics and computers. In addition, by shifting the pitch gauge over the tooth width an impression of the deviations in the tooth trace can be gained. The pitch gauge was developed at the time the DOT gears were built.

During the assembly at Svenska Bearing AB the line contact was tested by spraying a thin layer of Molykote G-Rapid plus on the teeth, such that it was visible in the contact image, see Fig. 14. No adjustment of the pinion was required. This led to the conclusion that the large parts of the Turret were made very precisely by Svenska Bearing AB.

Molykote G-Rapid plus served at the same time as run-in paste. The molybdeene disulfide works in into the micro-ruggedness of the steel surfaces. It smoothenes the surfaces and minimizes the fluctuations in the friction afterwards.

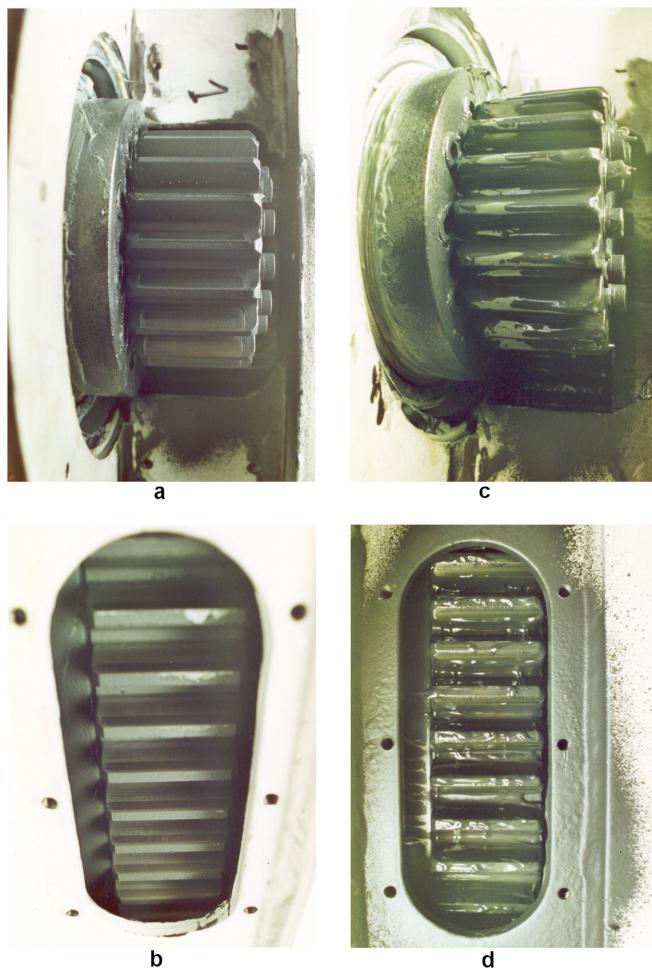


Fig. 14. Tests during the assembly at Svenska Bearing AB (see Fig. 4). Contact images (a) on the pinion, (b) on the large wheel. Visible is the contact over the whole width in the under part of the pinion and the corresponding upper tooth sides of the large wheel. On the other tooth sides the untouched thin film of Molykote G-Rapid plus still is present. After the running-in the teeth are greased with Molykote Longterm 00, a special gear grease. (c) on pinion, (d) on large wheel.

After the running-in, the teeth were covered with a grease layer of Molykote Longterm 00, see Fig. 14 and 15. This grease is pretty thin with the big advantage that, after being pressed away by the line contact between the teeth, it creeps back again over the whole tooth surface, of importance to the lubrication of the next tooth contact and to a good protection against corrosion. Especially this last point has a high value for the telescope, which is standing outside and is exposed to large temperature changes combined with weather situations of very penetrating wet clouds and rain pressed by heavy storms to the construction.

Next to the gears the encoder steel scale tapes of Heidenhain are placed. Originally, a double seal with two V-rings was proposed, see Fig. 5. At the end, a single large V-ring was used for the sake of a simpler construction. Practice learned that perhaps a further improvement is worth to implement. This can also be done afterwards with split rings to house and additional V-ring. The V-ring itself can be cut and glued for assembly. It was practised at the replacement of the V-rings of the tent-dome of the DOT. The original rings were made of Nitrile rubber, which turned out to be not stable against the solar light. They are replaced by Neoprene (Chlor) rubber rings which proved to be stable against both solar light and grease based on mineral oil.

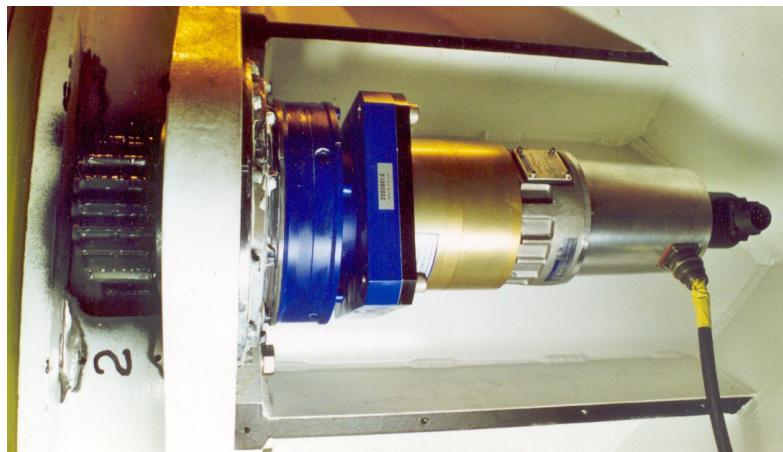


Fig. 15. Drive unit for the pinion. Visible are the components 9 to 13 of Fig. 5. The blue part is component 11, the planetary gear drive, $i=31$, from Alpha Getriebbau, which is suited for use from low speed to high speed due to its low friction. The torque required to move the slow shaft with unloaded fast shaft varies from 10 to 12 Nm for the 4 pieces delivered for the telescope.

The rubber seals of the large bearings received a special anti-friction treatment, which reduces the friction and particularly the fluctuations in it. The seals in the bearings are made of Nitrile rubber, because it is the most stable one against grease and it is the best one for the anti-friction treatment, although Neoprene rubber is also a possible choice. The described treatment is a surface treatment, which can be done afterwards. In our case it was done by Eriks Group NV (www.eriks.com).

The grease used in the large bearings is Molykote Longterm 2 plus. It belongs to the same family of greases as Longterm 00, but its consistency is much thicker, which makes it more suitable for roller bearings. Its oil viscosity is a little bit lower than that of Longterm 00. The grease paste used for the bolts, for the screw threads and the heads is Molykote DX. It gives a constant coefficient of friction and therefore is suited for tightening the bolts with a moment wrench to a prescribed pre-tension. It prevents stick-slip, also between bolted parts, when temperature differences give micro-motions between the parts.

All three greases are made up on the basis of lithium soap and mineral oil. Consequently, contact between these greases with a small scale local mixture does no harm. All three give an excellent corrosion protection and by combining these three greases a complete corrosion protection inside the construction is achieved.

The tooth width for the large gears is made 0.5 till 0.7 mm smaller than the nominal value. Consequently, there is enough clearance between the large gear wheels and the pinions to ensure that the gears will never clinch because of shrinking of the gear houses by fast cooling in the evening. Somewhat thinner teeth do no harm to the large gears; these teeth are strong and stiff enough anyhow. The limitations apply to the teeth of the pinions.

5. CONCLUSIONS

The stiff and low-friction construction of high precision bearings with incorporated gears in the compact design of the Turret is one of the contributions which have led to the success of the SST with new insights in solar physics.^{8,9} The high stiffness without stick-slip was confirmed by the tests with the control system.³ We have shown how this goal was reached with a systematic tackling of all influences on stiffness and friction, making compromises between several factors to gain a feasible production.

ACKNOWLEDGMENTS

We thank Bertil Petterson and Klas Bjelksjö of Stockholms Digital Mekanik AB for the fine cooperation, particularly Klas for the huge amount of design work. Thanks go to the co-workers of Svenska Bearing AB in Göteborg for the precise fabrication of the large parts and for their support during the test assembly.

Special thanks go to Ing. R. Marquardt of Hoesch Rothe Erde in Lippstadt and Dipl.-Ing. Gerhard Tripp of Flender in Bocholt. Mr. Marquardt was responsible for the bearing production with a remarkably high precision. He came with the ideas to realize the gear-teeth grinding after the completion of the bearings and to mount them on a thick plate. Mr. Tripp made the final gear calculations and conducted the fabrication and measurements. We thank Ing. Freddy Vasquez of Flender Nederland b.v. for his continuous help. We also thank the representative of Molykote in the Netherlands, MAVOM b.v. in Alphen a/d Rijn for the advise in questions concerning the lubrication and Eriks Alkmaar for the anti-friction treatment of the seals.

Special thanks go to Piet Hoogendoorn for the fabrication with high precision of the smaller gear parts, including the pinions, and for their assembly in Göteborg.

Special thanks also go to Rolf Kever for his indispensable and continuous help at La Palma.

Much of the applied gear technology originates from the development and construction of the Dutch Open Telescope (DOT) with a grant from the Technology Foundation STW.

The Swedish Solar Telescope (SST) is operated on the island of La Palma by the Royal Swedish Academy of Sciences at the Spanish Observatorio del Roque de los Muchachos of the Instituto de Astrofísica de Canarias.

REFERENCES

1. G.B. Scharmer, K. Bjelksjö, T. Korhonen, B. Lindberg and B. Petterson, "The 1-meter Swedish solar telescope", in *Innovative Telescopes and Instrumentation for Solar Astrophysics*, SPIE vol. 4853, pp. 341-350, 2002
2. G.B. Scharmer, P. Dettori, M.G. Löfdahl and M. Shand, "Adaptive optics system for the new Swedish solar telescope", in *Innovative Telescopes and Instrumentation for Solar Astrophysics*, SPIE vol. 4853, pp. 370-380, 2002
3. P. Dettori and G. Hosinsky, "The New Swedish Telescope Control System", in *Advanced Telescope and Instrumentation Control Software II*, SPIE vol. 4848, pp. 539-544, 2002
4. R.H. Hammerschlag, F.C.M. Bettonvil and A.P.L. Jägers, "Towers for telescopes with extreme stability, active or passive?", in *Astronomical Telescopes and Instrumentation*, SPIE vol. 6273, *Optomechanical Technologies for Astronomy*, paper 61, 2006
5. R.H. Hammerschlag, "A telescope drive with emphasis on stability", in *Advanced Technology Optical Telescopes II*, SPIE vol. 444, pp. 138-146, 1983
6. DIN-Taschenbuch: 106 Verzahnungsterminologie ISBN3-410-10840-8; 123 Zahnradfertigung ISBN3-410-10841-6; 173 Zahnradkonstruktion ISBN3-410-11426-2; Beuth Verlag GmbH, Berlin, Köln
7. MAAG-Taschenbuch: Berechnung und Herstellung von Verzahnungen in Theorie und Praxis. MAAG-Zahnräder-Aktiengesellschaft-Zürich, second edition, Zürich, 1985
8. G.B. Scharmer, B.V. Gudiksen, D. Kiselman, M.G. Löfdahl and L.H.M. Rouppe van der Voort, "Dark cores in sunspot penumbral filaments", *Nature*, 420, pp. 151-153, 2002
9. B. De Pontieu, R. Erdélyi and S.P. James, "Solar chromospheric spicules from the leakage of photospheric oscillations and flows", *Nature*, 430, pp. 536-539, 2004