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A telescope drive with emphasis on stability

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Abstract

A solar telescope, which consists of an open steel framework, is under construction. The telescope will operate without a dome in order to improve the local seeing. The telescope drives should be stable against the fluctuating wind forces. Calculations indicate that conventional gear drives show too much torsional and bending flexibility in the shaft of the pinion. In addition, the stability requires line contact between the teeth of meshing gears under a relatively low load, which requires an extremely good alignment of the gears. A new gear design is presented in which the pinion of the first gear stage and a pair of wheels of the second stage form a single block, which can rotate about a self-aligning spherical roller bearing. The line contacts between the teeth of the first and second gear stage, in combination with the spherical roller bearing, form a statically determinate system for this single block. In this way line contact between the gear stages is guaranteed and the single-block construction minimizes the torsional deflection. A shaft through the spherical roller bearing would bend too strongly. This problem is solved by a special shaft design, which incorporates the bearing in the shaft construction. The described design may be of interest for future large telescopes because it reduces the telescope vibrations caused by wind buffeting.

A telescope which needs uncommonly stable drives

Fluctuations in the air temperature in the direct neighborhood of a telescope deteriorate the image quality. In the case of solar observations, these fluctuations are strongly generated by the heating of the ground in the sun light. Wind will mix the air. Measurements of the temperature fluctuations showed, that a wind breeze of 5 - 10 m/s gives a rather homogeneous air temperature above a height of 15 m in the case of a flat ground surface. The solar telescope, which is under construction, will try to use the favorable seeing conditions, which are brought about by the wind. The telescope incorporates a tower support of 15 m height, which puts the optical instrument in the air with homogeneous temperature, as is illustrated in Figure 1. The tower and telescope itself should not disturb the air mass temperature around the telescope. To that end, the tower and as far as possible the telescope consist of open framework, which is transparent to the wind.

The tower construction is finished and is temporarily mounted for testing purposes on the site of the radio telescope at Westerbork, see Figure 2. The tower consists basically of 8 legs, which carry a platform, indicated by the letters N (North) and S (South) in Figure 1. The telescope itself will be mounted on three of the four corner points of the platform. The special geometrical configuration of the tower reduces the rotations of the platform caused by wind.[1] These rotations were measured on the Westerbork site with the help of optical interferometers. They were smaller than 0.1" in windgusts up to 9 m/s on top of the tower. The interferometric measurements also confirmed the mixing effect of the wind, which improves the seeing.[1]

The telescope on top of the platform in Figure 1 is shown enlarged in the schematic drawing of Figure 3. The primary mirror PM is placed above the declination axis DA. The open framework, which carries the box with secondary optics SO, permits the wind to blow through the main optical beam to and from the primary mirror. Stagnant air masses, which tend to get a deviating temperature, are avoided in this way. The mechanical construction of the telescope is dominated by the requirement, that the vibrations of the image due to mechanical vibrations in the telescope induced by the wind must be smaller than the optical resolution of 0.25" of the telescope. The stability of the framework parts is reached by proper application of some construction rules, which can be condensed in the principle "everywhere are triangles".[2]

How stability is reached in the drives

A first step to stable drives are a declination wheel DW (see Figure 3) and a right ascension wheel RW as large as possible. The wheels are incorporated in the framework constructions, which prevents the torsion of shafts. Two drives mesh in each large wheel to eliminate the backlash. In Figure 3 the two declination drives are indicated in side view near the letters DD. A sectional view of the first two gear pairs of one of the right ascension drives is indicated near the letters RD. The declination axis DA is formed by two spherical roller bearings, the right ascension axis by a spherical roller bearing SB and a roller raceway RR, which is supported by two rollers in the base frame on the tower platform.

At first, we tried to design the telescope with commercial available gear units. The thick shaft ends of the gear units were provided with pinions, which mesh in the declination wheel and in the right ascension wheel, respectively. Calculations showed, that the variable windload on the telescope would give too much torsional and bending deflection in the pinion shaft. In addition, a line contact between the meshing gears is required, because in case of a point contact only a part of the teeth face width contributes to the bending stiffness of the teeth and the local deformation under the Hertz pressure becomes substantial. (The first reason is more important for wheels with a small module in comparison with the face width. A small module allows the application of small pinions for high transmission ratios in compact drives and/or allows the application of pinions with many teeth for low frictional loss.)

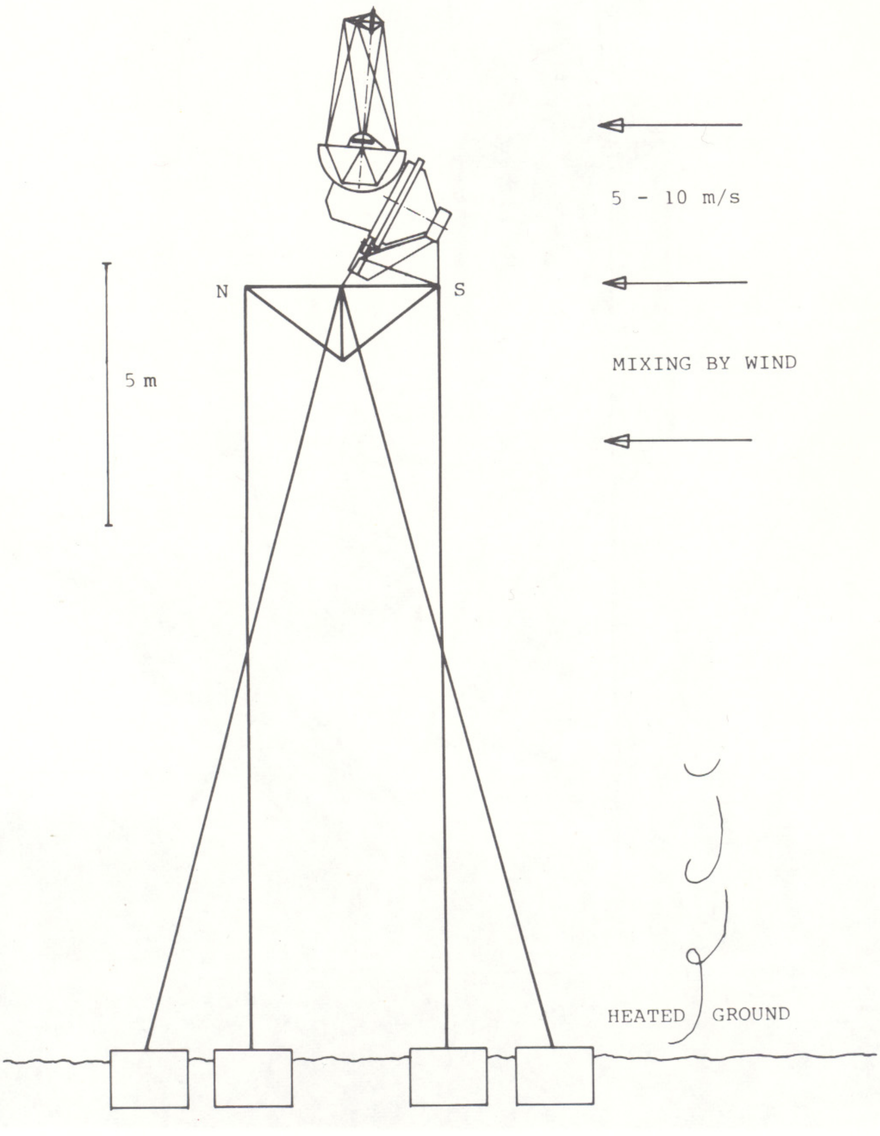

Figure 1. Principle of open tower telescope.

Line contacts between the pinions and respectively the declination wheel and the right ascension wheel would require an extremely good quality and alignment of the gears, of which the feasibility is doubtful with the means at our disposal. Moreover, for the gears inside the commercial gear units the line contact is reached by the deflections under heavy load. In the usual applications of gears the line contact does not serve the stiffness but it is only required for limitation of the material stresses in case of a heavy load. However, such a heavy load is not used for the telescope if the gear units have large enough dimensions to obtain the required stiffness, related to large enough face widths of the gears and shaft diameters. The inevitable conclusion was, that a special design is needed for the drives.

Figure 2. Tower temporarily mounted on the site of the Radio Telescope at Westerbork.

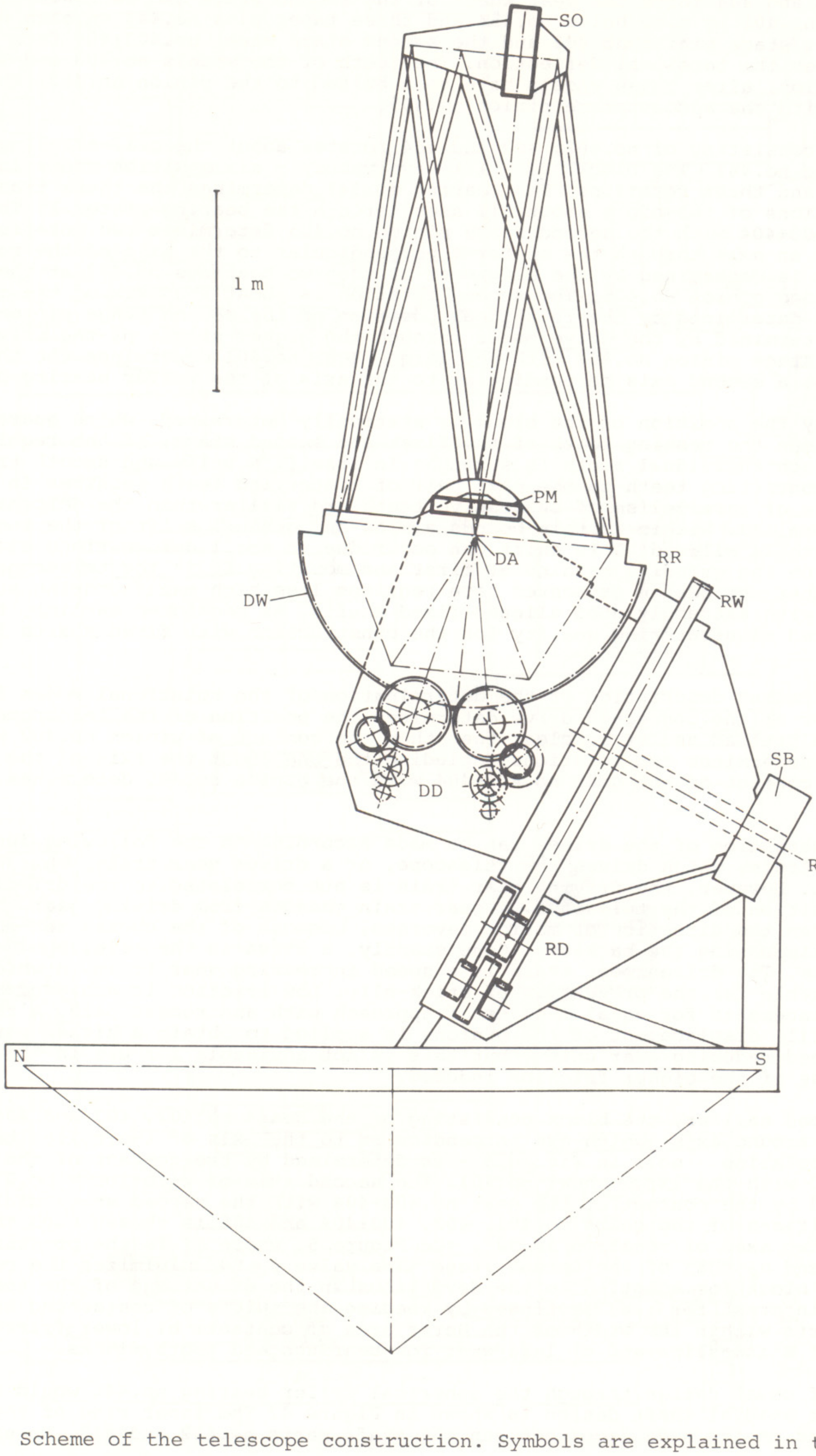

Figure 3. Scheme of the telescope construction. Symbols are explained in the text.

The developed design is shown in Figure 4. Involute cylindrical spur gears are used throughout the drives. The large gear wheel no.401, in this drawing the right ascension wheel, meshes with the pinion no.402. We call this the first stage of the drive. The pair of wheels no.403 and 404 forms the gear wheel of the second stage and is fixed to the first stage pinion no.402 by nine bolts no.454 and three taper pins no.448 on each side. In this way, the first stage pinion no.402 and the second stage wheel no.403+404 form a single block, which minimizes the torsional deflection. The teeth of the wheels no.403 and 404 are ground in one operation, after these wheels have been bolted to the pinion no.402. The wheel no.403 +404 meshes with the second stage pinion no.405.

The block consisting of no.402, 403 and 404 rotates about the self-aligning spherical roller bearing no.441. The block has - like every body - six position modes in space : three translations and three rotations. The bearing no.441 determines the three translations. It permits rotations of the block about all axes through the bearing center I. The contact of the gear no.403+404 with the second stage pinion no.405 determines two rotations. One rotation is about an axis through the center I perpendicular to the axis of the roller bearing no.441 and it is determined by the alignment of the two bearings no.472 at the shaft ends of the second stage pinion no.405. The second rotation is about the axis of the roller bearing no.441 and is determined by the rotational position of the second stage pinion no.405, which is in turn determined by the drive motor through the higher stages of the drive. The contact of the first stage pinion no.402 with the large wheel no.401 determines the third rotation, which is about a second axis perpendicular to the axis of the roller bearing no.441.

In this way the position of the block is statically determined, which guarantees line contacts between the meshing teeth of the first and second stage, if the requirement is fulfilled that each individual tooth is straight in itself. A stiff and smooth transition from one pair of contacting teeth to the next pair of contacting teeth requires in addition, that the deviations of parallelism of successive teeth are smaller than the deformations under the applied preload. Within certain bounds a slow and gradual shift of the tooth direction over many teeth is allowed. This shift can occur due to small deformations caused by mounting the gear on the grinding machine at first and mounting it in the telescope afterwards. This means, that the design presented here requires less high manufacturing standards than a construction with extremely good aligned fixed shafts. As mentioned earlier, the feasibility of the required manufacturing quality for the construction with fixed shafts is doubtful in this case.

In the preceding description of the determination of the rotational modes it was assumed, that the motor of the concerned drive determines the position of the telescope. The second drive gives a preload and it is clear that then the contact of pinion no.402 with the large wheel no.401 determines two rotations including the one about the axis of the bearing no.441, and that the contact of the gear no.403+404 with the pinion no.405 determines only one rotation.

An other partition of the drives can be made according to the following function : a driving gear train, which drives the telescope, or a driven gear train, which is driven by the telescope. However, the driving gear train is not correlated to the drive, which determines the position of the telescope. A gear train changes from driving gear train to driven gear train when the direction of motion reverses, because of the chosen set up with two drives for eliminating the backlash. Consequently, a drive in the telescope has alternately both functions. The driven gear train is a speed increasing gear train in which a low friction is desirable for the prevention of stick-slip. Low friction is achieved by balanced profile displacements for equal lengths of approach path and recess path. A relatively high positive profile displacement of the pinions is applied to obtain a higher permitted load in the usual speed reducing gear units, but this is not advisable for use in the telescope drives because of the higher friction induced.

As described earlier, the block consisting of the gears no.402, 403 and 404 has two rotational modes around axes, which are perpendicular to the axis of the roller bearing no.441. One axis of rotation - no.1 in Figure 5 - is determined by the contact of the first stage pinion no.402 with the large wheel no.401. The second axis of rotation - no.2 in Figure 5 - is determined by the contact of the gear no.403+404 with the second stage pinion no.405. The relative positions of the gears no.401, 402, 403+404 and 405 is chosen such that the angle μ between the two axes of rotation is 90°, see Figure 5, where α_1 is the pressure angle of the first stage and α_2 that of the second stage. The value $\mu = 90^{\circ}$ minimizes the correcting motions of the block for adaption to the deviations in the directions of the tooth traces, which is of interest for : a) stiffness by keeping the shifts of contacting tooth traces at their end parts within the width of the Hertz area of contact; b) lower friction for the prevention of stick-slip; and c) less wear for bearings and tooth flanks.

A shaft of usual design through the spherical roller bearing no.441 would bend too strongly. The special shaft design is shown in Figure 4. The inner ring of the bearing no. 441 is pressed on its side faces between the shaft parts no.412 and 413. On either side of

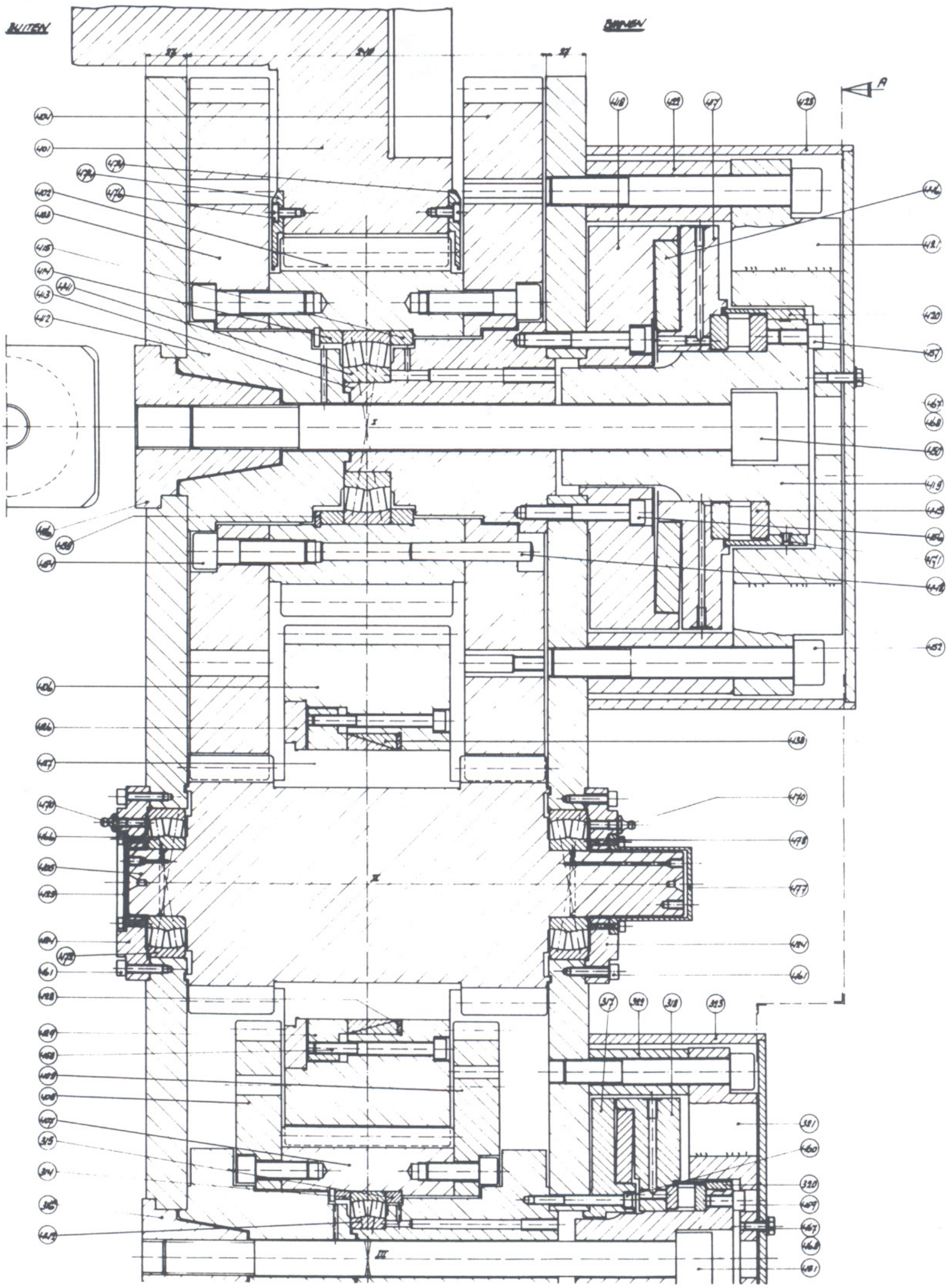

Figure 4. Design of the stable drive.

the bearing the shaft parts have diameters nearly equal to the outside diameter of the bearing. Between the shaft parts no.412 and 413 the inner ring of the bearing becomes also a shaft part, which contributes to the bending stiffness. The forces on the teeth of the gears no.402 and 403+404 cause a bending moment in the shaft. If fixed connections between the shaft parts no.412 and 413 and the inner ring of the bearing no.441 are assumed, then the maximum tensile stress in the connections can be calculated. The compressive stress caused by the preload on the shaft parts is made higher than the calculated maximum tensile stress. Hence, there remains always a compressive stress on the connections. The preload is adjusted with the hexagon socket head cap screw no.450. The disc spring no.446 allows for a smooth regulation and precise read out of the preload. The slit width between the disc spring shaft no.417 and disc spring house no.418 can be measured in the regions between the 4 spacer bushings no.422. The preload is calculated from the disc spring deflection, which follows from the measured slit width. A precise adjustment of the preload prevents unnecessarily high stresses in the inner ring of the bearing and in the screw. The roller thrust bearing no.445 reduces the required torque for turning screw no.450. A smaller torque serves also a precise adjustment of the preload.

In the case of fracture of screw no.450 the "bridge" part no.421 is a protection against parts, which are shot by the disc spring. The screw no.450 is turned with a key through the central hole in part no.421. Screw fracture is very unlikely, but without protection a fault in material of the screw no.450 could be fatal for the person, who turns it. The ten screws no.456 hold the shaft part no.413 in case of a failure of the screw no.450. Hence, after a failure the drive will be no longer of high stiffness, but the drive will not break down. The part no.421 is also used for locking the rotational position of the screw no.450 with the help of screw no.457. 9 holes are uniformly distributed over a circle in part no.421 and 8 tapped holes in part no.419. Hence, the part no.419 and with it, the screw no.450, can be locked in 72 positions of a revolution.

The third stage wheel no.406 is fixed on the second stage pinion no.405 with the help of the set of tapered rings no.438. The ring no.427 consists of two parts. In this way, the wheel no.406 can be mounted after the two sides with teeth of pinion no.405 are ground in one operation. The wheel no.406 meshes with the third stage pinion no.407. The construction of the third and fourth stage is similar to the first and second stage. The third stage

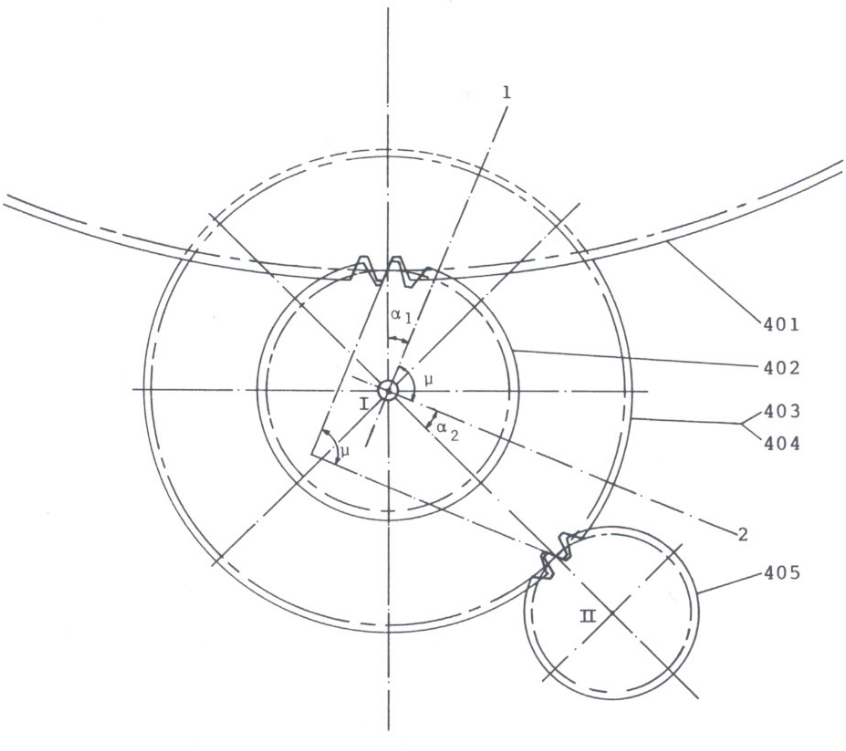

Figure 5. Schematic side view of the first two drive stages.

pinion no.407 and the fourth wheel no.408+409 form a single block, which rotates about the self-aligning spherical roller bearing no.442. The fourth wheel no.408+409 meshes with the fourth pinion no.410, see Figure 6.

Calculations show, that roughly the contribution of the flexibility of a gear stage to the overall flexibility of the telescope drive decreases quadratically with the transmission ratio between this stage and the first stage. Consequently, the stiffness becomes less important for the higher stages and in this case it is allowed to leave the gear box with a shaft end after the fourth stage. A schematic side view of the two declination drives, each with four stages, is shown near DD in Figure 3.

The tip surfaces of the declination wheel DW and right ascension wheel RW are covered with flat belts for protection against weather influences outside the drive housings. Inside the drive housings the belts are guided around the first stage pinions with the help of guide rollers. The sides of the toothing of the declination wheel and right ascension wheel are covered with the segments no.474, see Figure 4. Belt and segment form a closed protection shield for the toothing outside the drive housings.

The telescope is not balanced in order to obtain a construction with better seeing conditions and less wind sensitivity. In the case of an alt-azimuth mount there may be less need for an imbalance. But in this case a polar mount was chosen because of the simpler regulation system for the drives and the absence of image rotation. We expect, that the imbalance

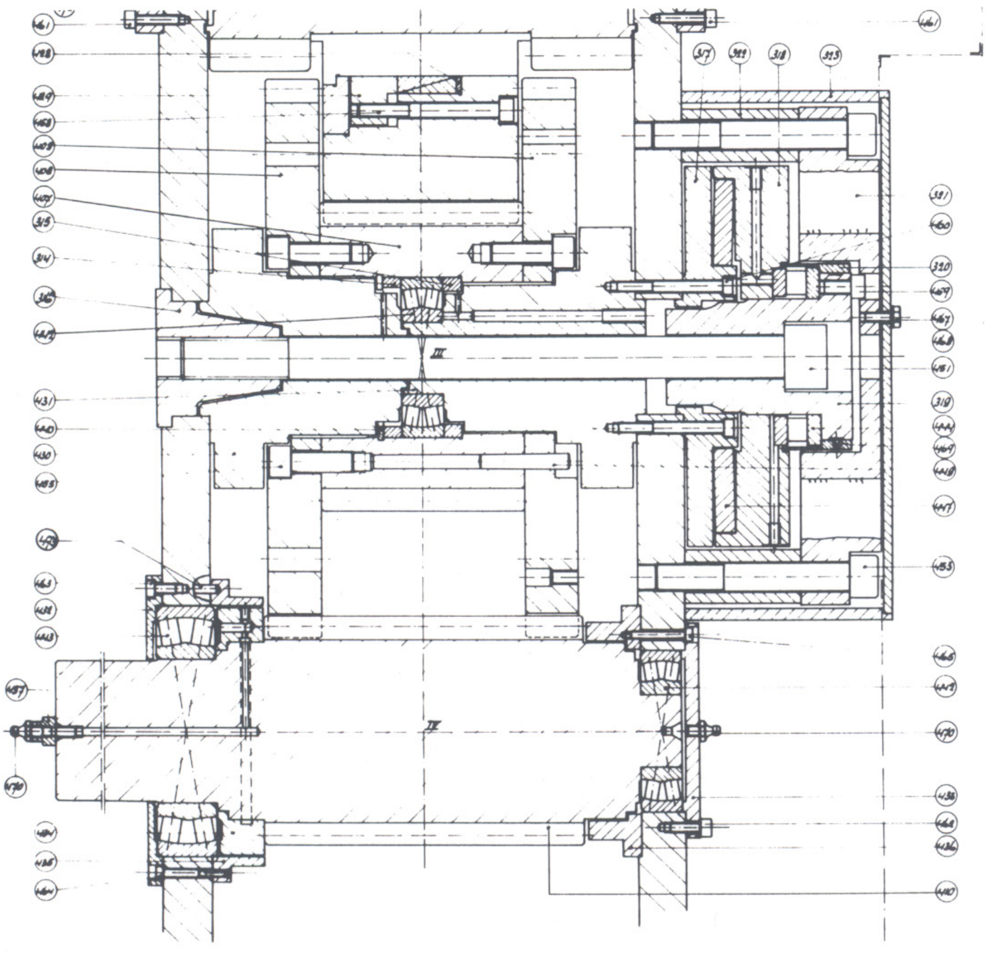

Figure 6. Fourth drive stage with shaft leaving the gear box.

is permitted, because the high drive stability reduces the risque of stick-slip. The pointing deviations by deflections under the imbalance load are limited by the high stability of the drives and the telescope frameworks, and a guiding system can correct for these pointing deviations. The imbalance can be reduced by counterweights afterwards, if our experiences would proove the necessity.

Roller friction drives implemented in all steel rollers can be stiffer than the described gear drives, because the roller drives miss the flexibility contribution due to the tooth deformation. However, the low coefficient of friction results in high contact forces. The desired permission for an imbalance was the main reason, that no roller friction drives have been chosen.

Conclusions

The described design shows a gear drive with low torsional and bending flexibility in the shafts and gear bodies. A systematic design based on statically determinate systems assures line contacts between the gears. Hence, the whole face width of the teeth contributes to the bending stiffness of the teeth and the local deformation under the Hertz pressure is minimized. The flexibility caused by the deformations of the teeth itself and of the direct neighborhood under the teeth in the gear bodies becomes the largest contribution to the flexibility of a gear stage. The contribution of a gear stage to the overall flexibility of the telescope drive decreases with the transmission ratio between this stage and the first stage. Hence, the flexibility of the teeth of the first stage settles the overall stiffness of the presented drive design.

Calculations show, that a further improvement of the stiffness is possible by the application of shorter teeth than those with the standard ratio of tooth depth to module. Shorter teeth can be applied, if the pinion has more teeth. There is a limit in the number of teeth of the pinion because the strength of the teeth or the tooth manufacturing requires a minimum module. We did not consider the use of shorter tooth profiles, because our (financial) means would not allow to experiment with non-standard tools for tooth generation.

The presented design is developed for the concept of open telescopes without wind protection by a dome. However, telescopes with a dome or other housing are still exposed to a certain windload, which increases with the size of the telescope. Hence, the described drive design may also be of interest for future large telescopes in housings.

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