Precise Low Cost Chain Gears for Heliostats

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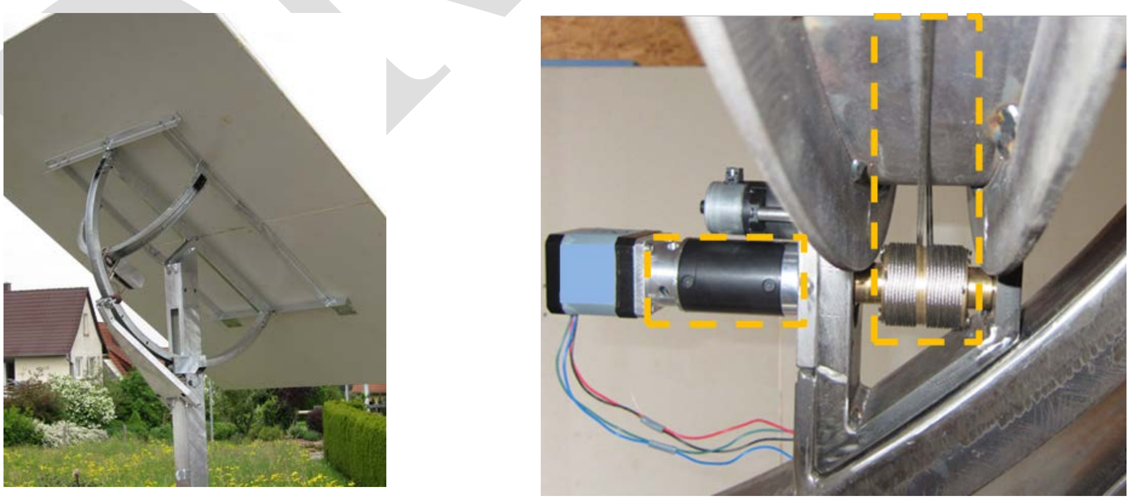
Abstract. This work investigates the potential of chain gears as precise and low cost driving systems for rim drive heliostats. After explaining chain gear basics the polygon effect and chain lengthening are investigated. The polygon effect could be measured by a heliostat with chain rim gear and the chain lengthening with an accordant test set up. Two gear stages are scope of this work: a rim gear and an intermediate gear. Dimensioning, pretensioning and designing for both stages are explained.

INTRODUCTION

Solar fields with heliostats represent the highest investment expenditures of a solar power tower plant. Therefore, investigation of different strategies for construction, operation and maintenance of heliostats offer high cost reduction potential. Central gear systems used in standard heliostat types must provide high tracking accuracy and solid construction to withstand wind and material loads. Big diameters of rim drive heliostats for long lever arms support precise tracking and lead to a reduction of loads [1]. A first prototype using a winch wheel cable drive system with an intermediate spur gear (Fig. 1) has been presented at SolarPACES 2014 conference in Beijing [2]. This work aims to reveal cost reduction potentials for rim drive heliostats using chain gears.



(a) (b)

FIGURE 1. (a): First rim cable drive prototype with 8 m² mirror surface; (b): Cable drive system with intermediate gear in the left and winch wheel in the right dashed box.

The intermediate spur gear with a gear ratio of 1:49 (Fig. 1, (b), left dashed box) is by far the most expensive component of the drive system. Therefore, a cheap alternative is wanted.

The winch wheel cable drive (Fig. 1 (b), right dashed box) is precise and of low cost. But mounting is somewhat tricky. Furthermore, for bigger heliostats and higher loads cables of bigger diameter would be required which would lead to a bigger diameter of the winch wheel which would reduce the gear reduction ratio. Alternatively, more cables of smaller diameter could be used, but this would complicate the mounting of the drive system which is already with two cables not trivial. Therefore, it is investigated whether also the cable drive could be replaced by a simpler system.

Several gear systems were analyzed and it could be concluded that for both gear stages chain gears are most promising regarding precision (if pretensioned) and low cost. Chain gears fit well to heliostats because rotational speed is low and maximum loads occur only at rare strong wind events. This offers low resonance speeds and excitation frequencies. Therefore, high lifetime can be expected (which was confirmed by a similar application [3]). For the azimuth drive of conventional heliostats a multi stage chain gear was already developed [4].

CHAIN GEAR BASICS

Chain gears are widely used in common applications such as bicycles, motorcycles as well as timing chains to operate poppet valves of camshafts in internal combustion engines with pistons. One major requirement of such drives for poppet valves usually is to provide high precision despite a high amount of rotations per minute.

By entering a rotating sprocket, the first contact of a roller (specified with “A” in Fig. 2) is at the flank of a tooth. During rotation, the rollers of a chain link unroll along the flank of a sprocket´s teeth. At the end of the rotating movement, the rollers have moved through the valley between two teeth and the rollers are in contact with the flank of the next tooth in rotating direction. While a sprocket represents an ideal round cycle, the chain cannot reproduce this cycle due to the stiffness of each chain link. Therefore, and to realize this shifting movement on all areas of tooth flank, a chain has a slightly larger distance between rollers of one chain link compared to the distance between two teeth along a sprocket (specified as “2α” in Fig. 2, also called pitch).

The precision of chain gears is realized by applying a tensile force in longitudinal direction of a chain that tightens the chain. During the rotation of a sprocket, this applied tension forces the rollers of a chain link into a specific position along the flanks of a tooth. The position varies depending on the angle of rotation and the limited options of a chain to reproduce an ideal cycle, depicted in Fig. 2 (dotted line). If tension is applied, the first teeth in contact with the rollers are exposed to the highest forces. The force on the other teeth reduces in rotational direction until the chain links lose contact with the sprocket (Fig. 2, bottom of the sprocket).

If a pretension is applied on both sides in longitudinal direction of the chain, the number of “active” chain links in contact with the sprocket can be increased. Furthermore, this offers a change of rotating directions without backlash, because tightening of the loose part of the chain as a first step is not necessary.

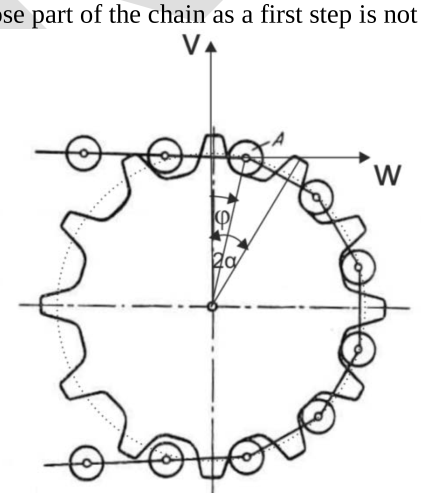


FIGURE 2. Example of different positions between chain rollers and sprocket teeth and angles for calculation of a polygon effect [5].

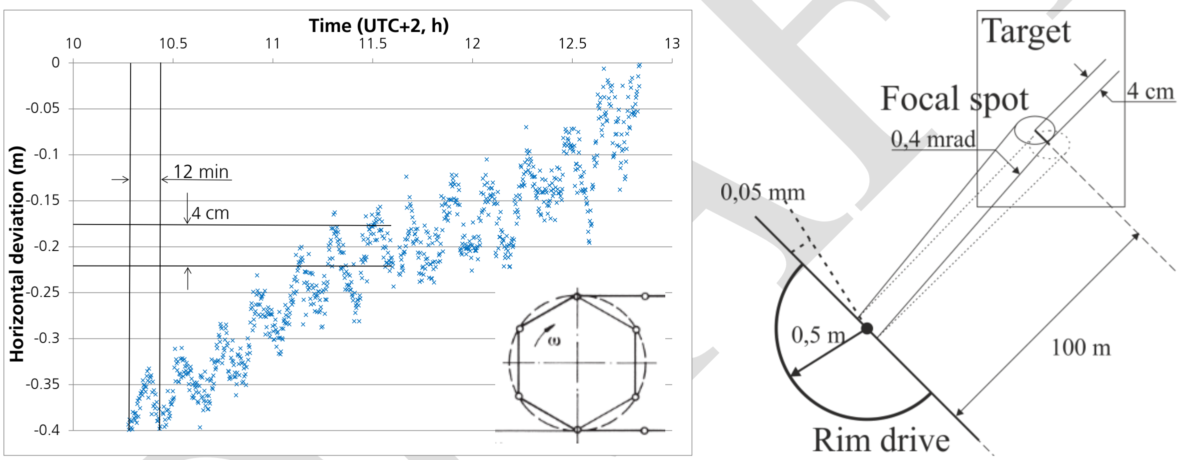
The polygon effect describes the difference between the length of the chain when bended around the sprocket (compare Fig. 2) and the length of an accordant perfect circle and the resulting differences in speed or acceleration.

Especially large chain pitches and small sprockets intensify the polygon effect which can be observed best during movement or under high load. The polygon effect needs to be considered especially for applications with high peripheral speed (because it will increase vibration of the system) and also to prevent an overload and increasing wear on single chain links due to the variations in length. The effect is specified by formula (1) and (2) by using the angles depicted in Fig. 1 [6]:

$$v = \Delta r = \frac{d}{2} \cdot (\cos\varphi - \cos\alpha) \tag{1}$$

$$w = \frac{d}{2} \cdot (\sin\varphi) \tag{2}$$

Formula (1) describes a fluctuating movement in v-direction (compare Fig. 2) to the sprocket. This is caused by the inability of the chain to follow an ideal cycle because of the limited numbers of chain links and the pitch angle α of the sprocket. The result is a deviation in v-direction. This leads to changing peripheral speeds during rotation, which can increase vibrations of the system. Formula (2) describes the tangential position (w) of the first roller of the chain in contact with the sprocket. The angle of the roller is specified as φ (Fig. 2). The effective lever arm of the chain varies because of the fluctuations in v-direction of the roller. The variation of the lever arm leads to a fluctuating speed of the roller (for constant rotational speed of the sprocket). This fluctuation leads to an additional fluctuating longitudinal force. Thus, for high pretension, the polygon effect may lead to high stress and energy consumption up to blocking of the system.



(a) (b)

FIGURE 3. (a): Fluctuating focal spot due to polygon effect. (b): Change of angle of drive and surface normal.

First measurements of tracking accuracy confirmed that the so called "polygon effect" has an impact on the movement of the focal spot (Fig. 3). The phase of the movement corresponds to the length of the single chain members and the amplitude to the calculated speed variation due to the polygon effect (according to equation (2)). In consideration of a constant rotational speed of the sprocket the deviation of the focal spot has local minima every 12 minutes with a mean deviation of ±2.0 cm or ±0.2 mrad. This is caused by a fluctuation of the surface normal of ±0.1 mrad. The periodic deviation of the focal spot caused by the polygon effect is not critical regarding tracking accuracy because it can be eliminated by the control. Only regarding live time the polygon effect would be critical if pretension is set too high. With a certain arrangement of the sprockets the polygon effect regarding chain lengthening can be even eliminated. This will be discussed in an additional paper (in preparation).

Although rotational speed is low, greasing and wear need to be considered for high lifetime. Wear of chain gears highly depend on regular greasing, the specifics of the oil, soiling and the material of the chain. In general, irregular and fragmentary lubrication as well as corrosion effects lead to higher wear. In addition, dry rotating chains allow about only 15 % of the designed surface pressure between rollers and bushes [7]. Wear of chains is usually represented by lengthening of the chain over time. With sufficient lubrication, wear of a chain occurs in different stages with degressive, linear and (after about 3 % lengthening of the chain) progressive wear. With insufficient lubrication, wear tends to be progressive, which decreases lifetime expectations [7].

RIM GEAR

Dimensioning

During stow the drives are locked and unloaded. Hence, for their dimensioning the wind loads during operation are decisive. They are calculated in the following according to [8] for an 8 m² heliostat with an aspect ratio (width to height) of 0.78. A typical maximum operational wind speed is 10 m/s at 10 m height. While the heliostats moves into stow position the wind could increase. Therefore a maximum wind speed of 15 m/s in 10 m height is assumed. For a wind profile according to the power law with an exponent of 0.15 the peak wind speed at first rotational axis height of 2.2 m is 12 m/s. With a gust factor of 1.6 the mean wind speed is 7.5 m/s. With the peak wind load coefficient for the hinge moment of 0.53 ([8] load case 2) and taking an air density of 1.25 kg/m³ and a safety factor of 1.15 into account this results in a maximum overturning moment during operation of 550 Nm about the first axis and 430 Nm about the second axis. As specified in Table 1, the diameter of the first rim is 1.4 m and of the second 1.0 m. Hence, the chain of the first rim is loaded at maximum by 790 N and the chain of the second rim by 860 N.

For the following rough life time estimation it is assumed that the mean loading is half the loading for maximum operational mean wind speed (which is a conservative assumption). According to [9] the mean wind load coefficient is 0.25 which leads to 260 Nm maximum mean wind loads for the first and to 200 Nm for the second axis. With the corresponding lever arms half of these loads lead to 190 N for the chain of the first rim and to 200 N for the second rim.

To evaluate the usage of bicycle chains for rim drive systems, load indices [7] are calculated and compared to bicycle chains under similar conditions. According to [10] and considering medium loads (100 W, 5% slope) for high lifetime expectations, an average tensile force of 210 N is estimated for bicycle chains.

A daily working angle for both axes of the rim drive heliostat was estimated. This angle considers the maximal tracking movement according to the sun angle as well as driving into a safe stow-position once a day to prevent damage by high wind loads.

Considering one circulation of the bicycle chain and the daily working angle of the rim chains, the estimated load indices for the rim gears are about 3 magnitudes less than for bicycle chains under comparable conditions. Therefore, bicycle chains can be a considered as adequate for rim drive chain gears. Nevertheless, industrial chains should be preferred due to better availability of validated data (e.g. joint surface, chain stiffness). Further investigations need to be carried out to prove this conclusion.

To calculate wear and tear, load collectives with 10 different wind speeds are considered (0-10 m/s, Weibull). The wind speed in height of the surface is calculated by [9] and the frequency distribution of wind speeds is considered by [11]. Furthermore, load moments are weighted by the frequency distribution and an operation with a dry chain was considered. With about 0.2 mm chain lengthening for 1^{st} rim and 2^{nd} rim, the estimated wear is low (details of the calculations to be published soon). Chain housings are considered to reduce friction loss by reducing loss of lubrication and protection against soiling.

TABLE 1. Estimation of load indices and wear for rim gear compared with typical usage of a chain gear.

Parameter	Rim gear	Bicycle chain
Daily working angle (1^{st} rim / 2^{nd} rim)	230 / 280 °	(chain length: 1.5 m)
Max. allowed wind load for operation (10 m height)	550 / 430 Nm	-
Radius / lever arm (1^{st} rim / 2^{nd} rim)	0.7 / 0.5 m	-
Max. allowed tensile force (1^{st} rim / 2^{nd} rim)	790 / 860 N	-
Average tensile force (1^{st} rim / 2^{nd} rim) (2.2 m height)	190 / 200 N	210 N
Estimated load indices (1^{st} rim / 2^{nd} rim)	1.4E-06 / 1.8E-06	4.884E-03
Wear / chain lengthening (1^{st} rim and 2^{nd} rim)	0.2 mm	-

Pretensioning

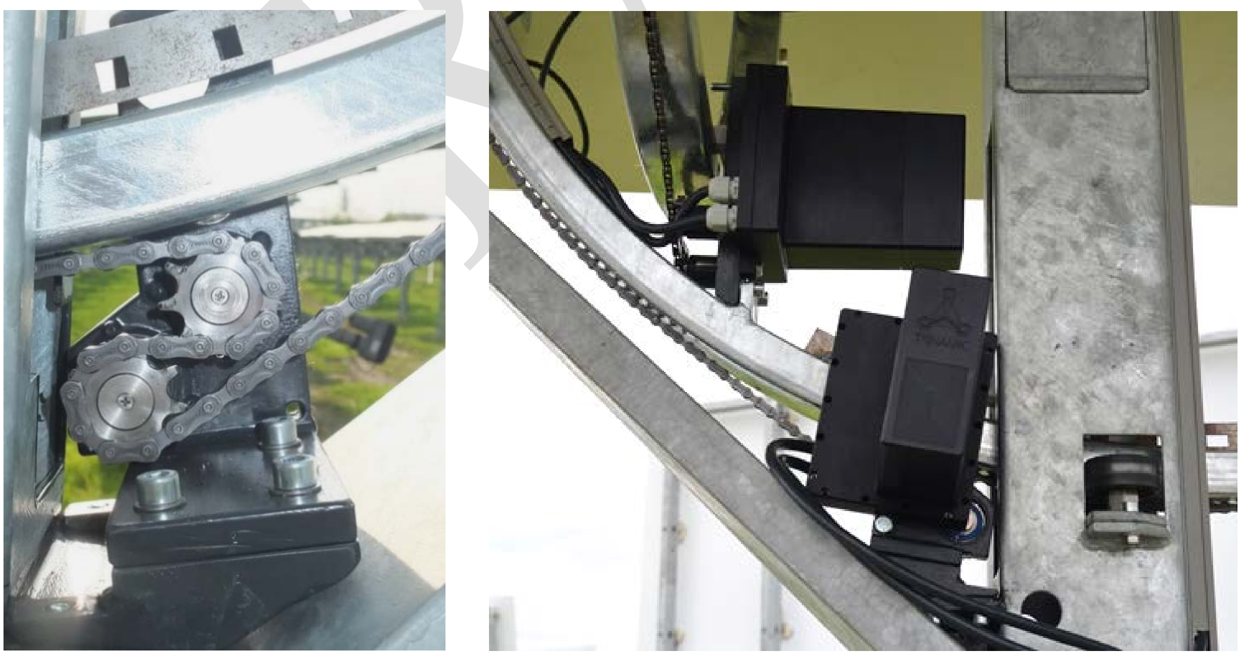
The pretensioning of chain gears is needed to provide high precision and low backlash. The tension must withstand the maximum wind loads acceptable in operation mode. In situations where the operational wind speed is exceeded, the heliostat moves to a stow position with horizontal mirror facet. In stow, the loads are transferred through a locking mechanism to prevent overload and damage of the chain gear system.

(a) (b)

FIGURE 3. (a): Pretensioning in the rim gear stage by compressing disc springs with nuts. (b): Arrangement of rim chains with pretension mechanism

For the rim gear stage pretension is realized by compressing disc springs at the mounting on the rim, depicted in Fig. 3. Nuts can be easily turned to adjust the tension. First tests showed that due to the pretensioning the heliostat can be moved backlash-free. Too high pretensioning must be avoided because it would additional stress in the chain due to the polygon effect, as expected.

Design



(a) (b)

FIGURE 4. (a): "Z"-arrangement of sprockets for 1st rim; (b): Mounting of the drives.

The designs of chain rim drives for a 8 m² heliostat are presented in Fig. 4. To provide high arc of contact, the chain wheel drive uses a “Z”-arrangement for the sprockets (Fig. 4 (a)). Because of more rollers in contact with the sprocket, the risk of backlash due to wear or lengthening of the chain over time is reduced. The upper sprocket is connected with the intermediate gear and the lower sprocket is used to increase the arc of contact.

Test Setup

(a)

(b)

FIGURE 5. (a): First rim drive prototype with 8 m² mirror surface mounted at Solar Tower Jülich, Germany (without drive units); (b): Detail.

A first rim drive prototype has been mounted at the Solar Tower Jülich as a test-setup for tracking accuracy, energy consumption and software calibration (Fig. 5). Each rim is driven by equal drive units which include a mounting plate, two sprockets, an intermediate gear and an electric engine. Each chain is mounted at the bottom of the rims and set under pretension to increase accuracy. First tests showed functioning of the drive systems and tracking algorithms. Long term test are in preparation.

INTERMEDIATE GEAR

Dimensioning

In addition to the rim gear, further calculations for dimensioning a chain gear as replacement of the intermediate gear were made. As shown in Table 2, a similar method as for rim gears was used to calculate the load indices and wear. The maximal allowed induced torque is calculated considering the wind loads on the heliostat and the gear ratio of the rim drives while neglecting forces induced due to the polygon effect.

TABLE 2. Estimation of load indices and wear for intermediate gear compared with typical usage of a chain gear.

Parameter	Intermediate gear	Bicycle chain
Max. induced torque (due to max. wind loads)	10 Nm	-
Radius of large sprocket	0.12 m	-
Average load	83 N	(100 N)
Estimated load indices	6.0E-06	3.9E-03
Wear / chain lengthening	0.38 mm	-

Further estimations resulted in a decision for a gear ratio of 1:10 with one gear stage. A larger gear ratio with one gear stage could be chosen, but would lead to larger sprockets, which are more expensive and heavier. It would also increase the risk of skipping a tooth on the smaller sprocket by lengthening of a chain due to wear.

Using more than one gear stage would increase the number and the costs of the components too (sprockets, bearings, shafts), as well as the complexity and weight of the gear, but reduce the demand of the electrical engine, as listed in Table 3. Nevertheless, the lower costs of one gear stage with more powerful engine were preferred. If 8 teeth for the small sprocket turn out to be too small, a variant with 9 teeth could be chosen. But the increase in weight and effort for the big sprocket would be significant.

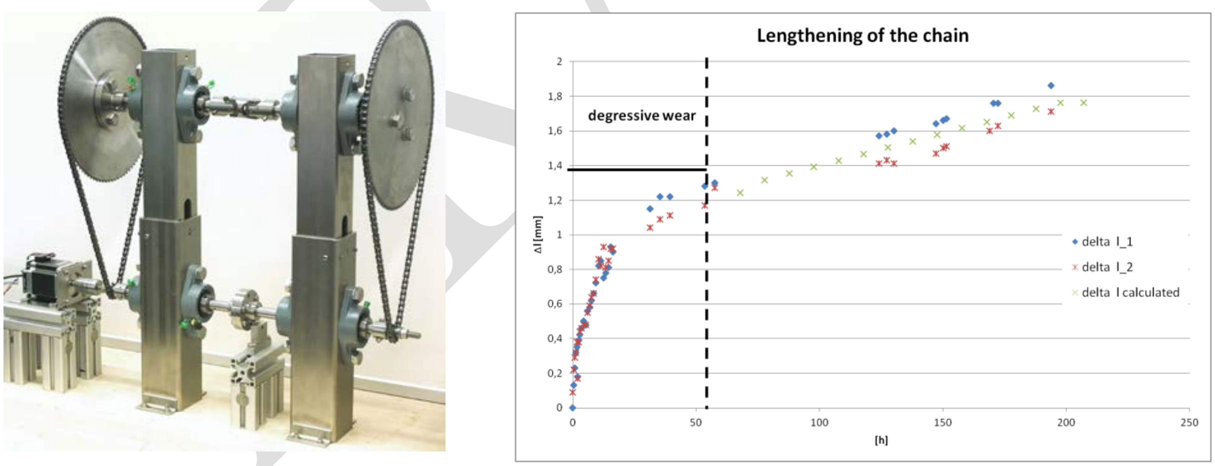
TABLE 3. Specifications of different stages for intermediate gear

Parameter	1 stage gear	2 stage gear
Number of teeth (small / big sprocket)	Var. 1: 8 / 80 (Var. 2: 9 / 95)	9 / 65
Torque demand at electrical engine	Var. 1: 1.8 Nm (Var. 2: 1.7 Nm)	0.42 Nm
Sum of weight for sprockets (according to the weight of commercially available sprockets)	Var. 1: 946 g (Var. 2: 1346 g)	1252 g

Pretensioning

Pretension in chain gears is usually realized by a mechanism of another sprocket connected to a spring which pushes the chain in radial direction by inducing a continuous force. The spring offers the possibility to apply a defined pretension whilst compensating lengthening of the chain due to wear and tear. But additional parts are needed which increase cost. Another way to realize pretension is to increase the distance between the two sprockets, for example realized by a telescope mechanism. By extending such a telescope mechanism, pretension is applied to the chain. The position of the guiding bar could be fixed by grub screws. For the chain gear test setup pretensioning is realized in this way (Fig. 6 (a)).

Test Setup



(a) (b)

FIGURE 6. (a): Scheme of the setup for longtime tests of intermediate chain gear. (b): Lengthening of the chain

The loads on chain gears for testing can be applied in different ways, for example by an engine to drive a system or by brakes to control or reduce rotational speed. There are different options available for brakes: disc brakes, drum brakes etc. To investigate functioning and behavior under load, both options have disadvantages if specific loads have to be applied. In the case of the heliostat in this work, loads are caused by wind and the weight of the components. A test rig was built to perform long-time tests, prove functionality and investigate wear and tear under load. It consists of two chains coupled by a clutch and cardan shaft. This arrangement provides a closed power circuit of two similar loops. The big sprockets are mounted on a telescope guiding bar and connected by a cardan shaft as well as with chains that are connected with smaller sprockets on the lower shaft. These two vertical arranged loops of chains use similar components (chains and sprockets). As depicted in Fig. 6, the electrical engine is mounted on the bottom of one loop.

The test setup is designed to analyze the relation between wear, pretension and efficiency over time. For this, the chains have to be loaded. This is done by three steps: fixing one of the two chain loops, inducing a specific torque in the other loop and then connecting both loops with a clutch. The torque is applied by weights that are connected at a sprocket of the non-fixed chain loop. The lever arm between the shaft and the mounting of the weight at the sprocket is considered for torque calculation. By using different weights, different load conditions can be investigated. First tests of wear show typical process of lengthening of the two chains (Fig. 6 (b)). With surface pressure of 2400 N/cm² for stage first (degressive wear) and 1450 N/cm² for the second stage (linear wear) and simulating 20 years of operation the measured wear of the chains is according to the calculated wear. These first tests were performed with dry chains to get more pronounced results. Greasing would of course reduce wear significantly.

CONCLUSIONS

By this work the high potential for cost reduction of chain gears for rim drive heliostats could be shown. Former work on winch wheel drive systems showed complex mounting, low up-scale potential for larger heliostats and high cost (for the intermediate gear). To be able to avoid these disadvantages, chain gears for the rim stage and the intermediate stage were investigated. Chain gears are precise when pretensioned by applying a longitudinal tension to the chain. It was shown that long life time can be expected due to short working distances of the chains, low number of alternating loads and pivot angles as well as low friction losses compared to chain gears used in bicycles.

The rim gear is designed with a “Z”-arrangement of two small sprockets to increase the arc of contact along the sprockets. Continuous pretension is realized in a simple way by compressing disc springs at the end of the chain. First tests with a rim drive prototype proved the concept.

The intermediate chain gear was designed with one gear stage to reduce overall costs, weight of components and risk of skipping a tooth due to lengthening of the chain. Pretension is realized by a low cost telescope mechanism to lengthen the chain. A test setup was designed to analyze the relation between wear, pretension and efficiency over time. First tests validated the calculated lengthening of the chains.

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