SPECIÁLIS SEBESSÉGVÁLTÓ TERVEZÉSE

DESIGN OF SPECIAL GEARBOX

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ABSTRACT
In general, it is specifically significant to design the units of the Pneumobile race vehicles according to the competition's principles. It could mean that we do the dimensioning of the parts for a short lifetime in order to save mass, but also to implement everyday solutions particularly for the needs of the competition to minimize costs. Nevertheless, a present-day Pneumobile is always a small moving test lab, where the students can try out such own ideas which help to improve their professional development. This paper presents the design and dimensioning of such a gearbox, which was created for a Pneumobile. This gearbox belongs to the orbital gear-drive systems. This kind of solution is almost only used in toys nowadays, but it will be clearly seen that this is suitable for this usage as well. As compared to the well-known applications there is a fundamental difference that we want to switch the gearbox under load without a disengaging clutch or stopping the motor. While connecting the gears, the load of the teeth can be increased to the multiples of the nominal load, which has to be handled and taken into account by dimensioning. For this reason, we present in this paper the basic construction and operation of the gearbox as well as those factors that we considered during the design and dimensioning. Currently we are testing the second version of the gearbox, thus we share the gained experiences from the first version as well.

Definitions:
Pneumobile: a vehicle, which transforms the energy of the pressurized gases with the help of pneumatic parts into kinetic energy [1].

1.INTRODUCTION

Pneumobiles at the University of Miskolc

The teams of the University of Miskolc are being taken part in the Pneumobile competitions since 2008. One team is typically at work for 3-4 years. In their first year, the teams usually aim to build functionally working vehicles, but later they develop these Pneumobiles in several way. [2], [3], [4].

For the second or third year, a Pneumobile is a moving test lab, where the students try out their own different ideas.

2. REQUIREMENTS FOR A GEARBOX OF A PNEUMOBILE

The need of a gearbox in the Pneumobiles are not certainly necessary thanks to the design and operation of the motors. However, it could be an advantage by deeper analyzing of the processes in the cylinders, if the transmission ratio can be modified during operation. In addition to these, the competition firstly provides the professional development of the students. From this aspect, designing any kind of mechanical unit can be beneficial.

In this paper, we present the design of a given Pneumobile's gearbox, therefore the requirements are valid for this vehicle.

- The maximum RPM of the given vehicle's driven shaft is 342 1/min, the torque on the driven shaft is 42 Nm.
- The gearbox must have two forward gears.
- One of the gear's transmission ratio must be underdrive, the other must be overdrive. Their proportion must be at least 2.
- The gearbox must have a neutral.
- Connection of the gears during the operation of the motor without a clutch is an expectation.
- The parts of the gearbox must be manufactured on general machining tools.
- For cost effective reasons, straight-toothed gears with the module of 1, 1.5, 2 or 3mm must be used.

Based the above written requirements, the design of an orbital gearbox seemed reasonable.

3. THE OPERATION OF THE GEARBOX

Fundamentally, this sort of a gearbox is built up by four pinions with the same number of teeth and two other toothed gears with different number of teeth from the before mentioned pinions and from each other as well. The operation principle of the gearbox is based on a rotatable bearing seated console (central unit), between the continuously rotating drive- and driven-shafts. Thus, in the console the reciprocal of the transmission ratio can be generated without using additional parts (Figure 1.).

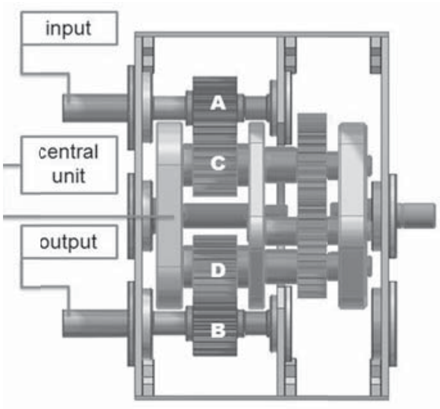

Figure 1: The Structure of the Gearbox

The geometrical design of the toothed-gears were done according to the DIN 3960 standard. The toothing of the gears are straight outside and pressure angle is 20°.

The module and number of teeth, so the dimensions and the centre distance of the gears were determined by cost effectiveness and simplicity. It means that the available bearing sizes on the market in a given budget and the necessary parts for the installation define how big the centre distances could be. Based on the centre distances, the size of module was chosen for the possibly biggest in a way that addendum modification is not needed. While defining the number of teeth it was considered that the contact ratio factor must be above 1,2. The minimum lifetime of the gears and bearings was specified as 5 operating hours. By determination of the lubrication factor, the opened gearbox and lubricated operation were taken into account.

On the grounds of the figures 2 and 3 it is easy to understand the operation of gearbox. On the drive- and driven-shafts one identical piece of pinion (Fig. 1: A and B) with 20 number of teeth are placed. A same pinion (C in the Fig. 1.) with 20 number of teeth on the central unit is connected to the pinion of the drive-shaft in the 1st gear, which turns on a common shaft with a toothed gear with X number of teeth. This toothed gear with X number of teeth is connected a toothed gear with Y number of teeth. The toothed gear with Y number of teeth is placed on a common shaft with a pinion (Figure 1: D) with 20 number of teeth, which is connected to the pinion with 20 number of teeth of the driven-shaft. In this state the transmission ratio of the gearbox is X:Y. If the central unit if turned by 90°, there is no connection between the drive- and driven-shaft. This is the neutral of the gearbox. If the central unit is turned by 90° again, the drive- and driven are connected again. This is the 2nd gear of the gearbox, where the transmission ratio is Y:X. In case of a further turn by 90° neutral comes again (Figure 2. and 3.).

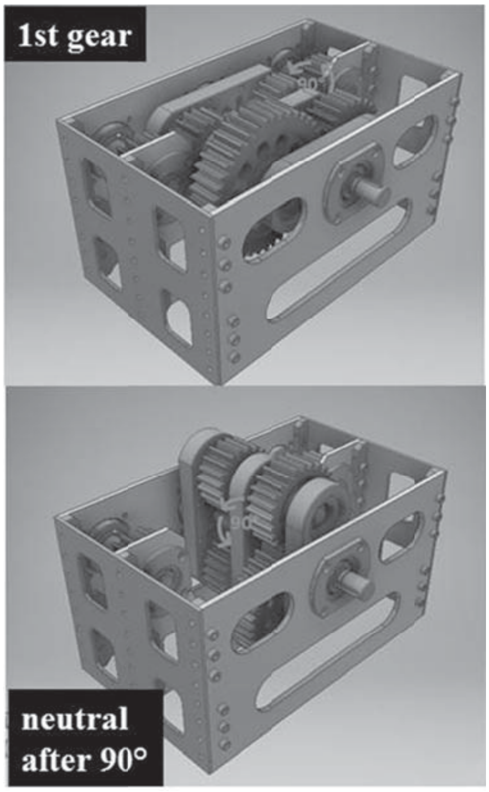

Figure 2: The 1st Gear and Neutral

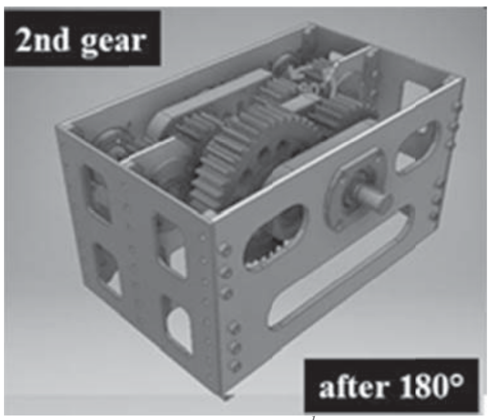

Figure 3: The 2nd Gear

The turning motion can be realized by many mechanisms, but automation can be done simply as well. We have chosen the manual coupling. Coupling of the gears can be adjusted easily to the needs of the driver. A coupling can be worked out, when an actuating arm needs to be moved in opposite direction or another one, which needs to be moved only in one direction and the arm returns to its initial position between the gears.

In our solution for setting and adjusting certain positions a Geneva drive is provided. This can guarantee the necessary discreet angular displacement for the right connection of the pinions and the fixing of the central unit in connected gears in the same time (Figure 4).

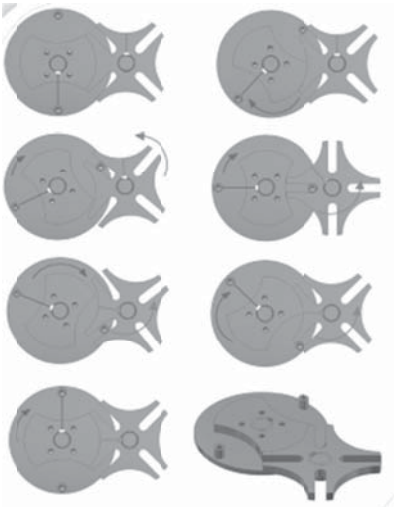

Figure 4: The Operation of a Geneva Drive

There is one additional positive advantage of the Geneva drive that all of the drive's parts can be simply manufactured on lathe and milling cutter, so we can make the drive easily.

4. SPECIAL DIMENSIONING PRINCIPALS OF THE GEAR-DRIVES

The control of the strength calculation of the toothed gears and pinions were done according to the DIN 3990. Since their geometrical dimensions are basically determined by other parameters of the gearbox, the toothed gears and pinions are significantly bigger than the given loads and the expected lifetime would require. The strength calculation were done according to the DIN 3990:1987 B [11]. The material of the toothed gears and pinions is 16MnCr5 and the surface treatment is nitridation, because its costs are favorable. The value of the K_A operating factor was chosen for 1.75, because the drive is under brusty loads while connection the gears and the direction change of the cylinder. By the calculation the parameters of the grease was fit for the UNIGEAR LA02 grease. Based on the calculations, the pinions of the drive- and driven shafts (A and B) and the pinions of the central unit (C and D) are used the most. Their safety factor of the tooth root strength is 3.06 and the safety factor of the flank pressure is 2.08. This last value may seem small, but the maximum value of the motor's torque loads teeth only for a short time because of the operating characteristics of the motor. The load is much smaller approximately in 90 percent of the operation.

After all we experienced during the race that the teeth were significantly damaged on all of the four pinions (A,B,C and D).

Firstly the root cause was searched in the failure of the surface treatment, but was found in the specialty of the gearbox's design. The central unit of the first gearbox can be seen on the Figure 5. It means that the A and B pinions turn in the opposite direction as the C and D pinions (Figure 1.). This has the results while coupling that the central unit wants to turn out from the connection and it makes the turnover harder, so the coupling process is uncertain. The teeth are loaded with huge dynamic impacts depending on velocity of the vehicle and the RPM of the motor. This fact was taken into consideration with value of the K_A factor, but we have chosen 1.75, because these loads act for a short period.

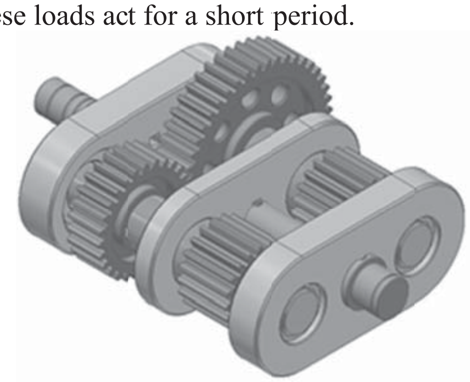

Figure 5: The First Design of the Central Unit

For the redesign of the gearbox the right values of the parameters had to be defined, so the root causes of the failure had to be revealed.

5. ROOT CAUSES OF THE FAILURE

The calculations were done only for the states of the connected gears of the gearbox. According to the experiments, it was determined that the coupling process takes place for at least 1 second. During this time the pinions rotate 5.7 turns. In these moments the center distance changes and the contact ratio factor is smaller than the nominal. As a first step, we have searched that point where value of the contact ratio factor is already not zero.

This could be easily done by a graphic method (Figures 6 – 9). This state takes place when the center distance of the A and C or A and D pinions are 44mm. The contact ratio factor is 0.033 at this time (Figure 6).

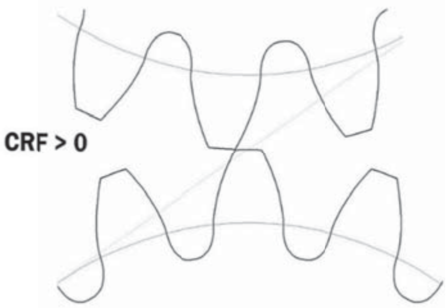

Figure 6: Modelling of the Effect of Centre Distance on the Contact Ratio Factor
A_w: 44mm.

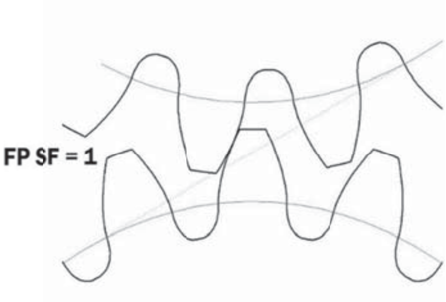

Figure 7: Modelling of the Effect of Centre Distance on the Contact Ratio Factor
A_w: 42,38mm.

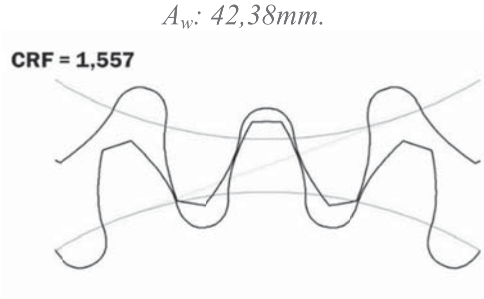

Figure 9: Modelling of the Effect of Centre Distance on the Contact Ratio Factor
A_w: 40mm.

In this way a range can be defined where a certain point has to be found when the safety factor of the flank pressure is 1. At such points the center distance is 42.38mm and the contact ratio factor is 0.625 (Figure 7).

The next wanted point is that when the contact ratio factor is 1. For this point belongs 41.05mm of center distance (Figure 8).

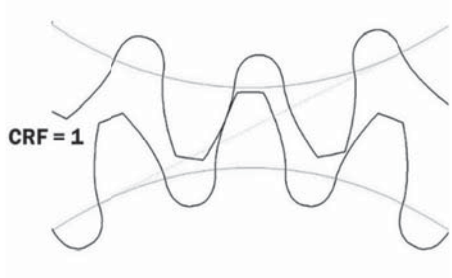

Figure 8: Modelling of the Effect of Centre Distance on the Contact Ratio factor
A_w: 41,05mm.

The Figure 9 shows the state of full connection. The contact ratio factor is 1,557.

It means that the pinions rotate ca. 4 turns while coupling besides the contact ratio factor of 1, the safety factor of the flank pressure is under 1 during more than the one third of the coupling time. The contact ratio factor under 1 means itself as well that the operation factor was chosen wrongly. The period of time when the surface of the teeth is under huge flank pressure was underestimated.

Besides this we did not consider in the lubrication factor that one part of the grease does not stay on the surface of the teeth and dust sticks into the grease because of the open design, so the lubrication factor is not true for the whole expected lifetime.

6. THE POSSIBLE SOLUTIONS

For the problem of the flank pressure among industrial circumstances a material with bigger strength and more resistant surface would be a good answer, which can be provided e.g. by case-hardening. This solution could not be chosen because of the costs.

Increasing the connecting surfaces is also a good solution. We did not want to modify the bearing seats and the frame of the gearbox, because this would have gone with significant costs.

The safety of the coupling had to be increased. This was important from the aspect of decreasing the coupling time and from the aspect of the usage as well. The most practical solution for these was to modify the direction of rotation of the toothed gears in the central unit.

6.1 Considered other parameters at redesign
The lubrication had to be re-evaluated, the value of the lubrication factor was modified. The new value was set according to the research experiences from the field of the open gear drives operated without lubrication in the Institute of Machine and Product Design at the University of Miskolc [6], [7].

The central unit moves a bit during operation because of the manufacturing inaccuracy of the Geneva drive, therefore the center distance is not constant. These limits were measured and the most unfavorable state was taken into account at the calculations.

7. THE SOLUTIONS
We investigated how the direction of rotation at the toothed gears in the central unit could be modified. The best solution was to implement additional pinions.

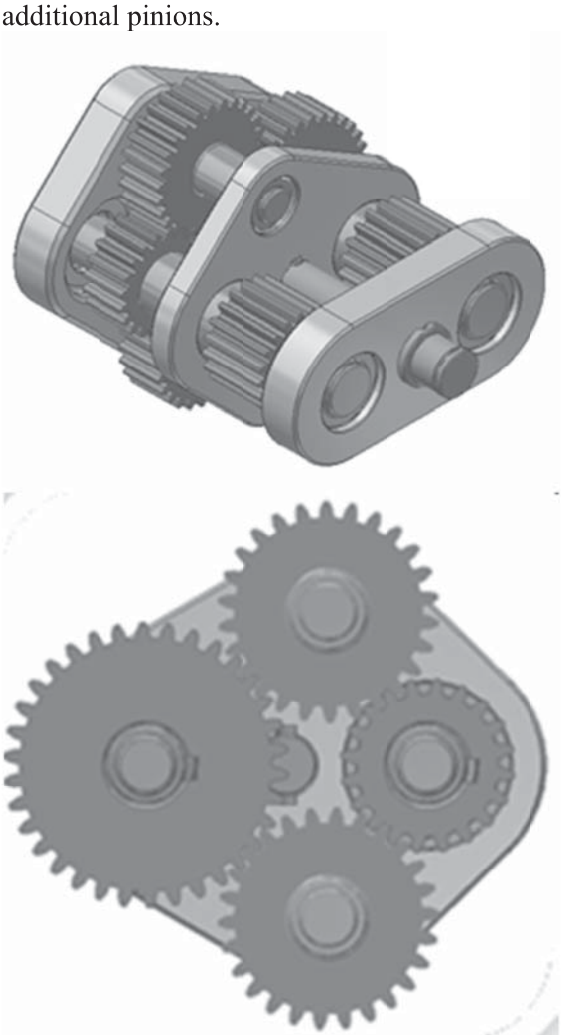

Figure 10: The Modified Central Unit

As shown in the Figure 13, the pinions are now rotating in the same direction.

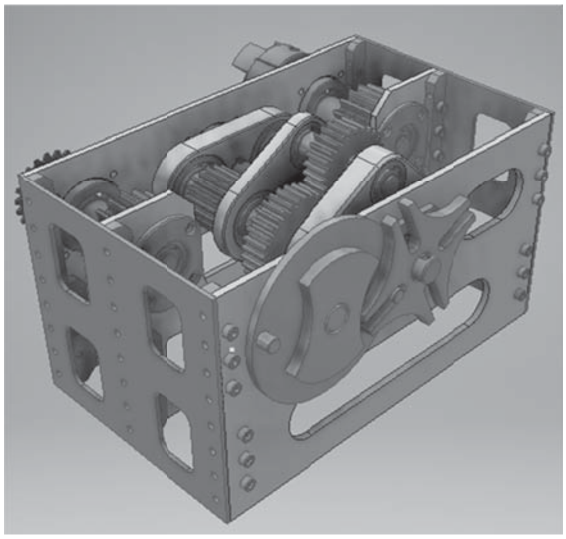

Figure 11: The New Central Unit Fits in the Old Housing

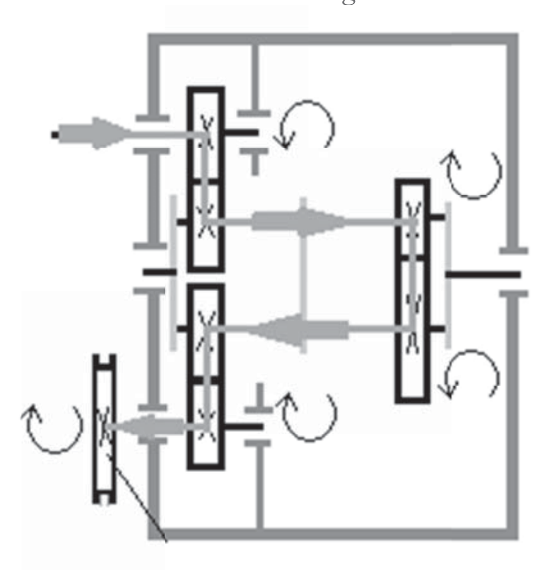

Figure 12: Kinematic Sketch of the Old Design

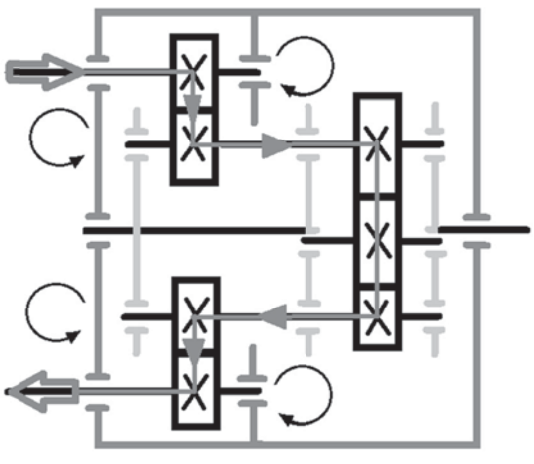

Figure 13: kinematic sketch of the new design

Thereafter, we examined that how big module and face width are necessary with the new parameters in order to have a sufficient flank pressure at least in 70 percent of the coupling time.

For this calculation, the value of the lubrication factor was chosen for 0.4 and the value of the K_A operating factor was increased to 2.25. From the available modules with fair prices the 3mm was chosen and after investigating the available space we increased the face width to 30mm.

In this manner, we got 6.72 for the safety factor of the flank pressure, which does not decrease to a critical value neither in case of a very small contact ratio factor.

This fact, that the centre distance is not constant means also, that it does not makes sense to specify an ISO 1328 accuracy grade 5 for the pinions, the accuracy grade 8 may be sufficient [8], [9]. This means significantly less cost, which allows the pinions to be manufactured of hardened BC3 steel.

8. SUMMARY

Penumobile competitions are perfect opportunities to test ideas of students in the field, they can push their boundaries. One of those boundaries is the design of complete working gearboxes. Our gearbox is working fine now, has reached its calculated lifespan without any signs of gear damage.

The contact ratio factor is one of the most important parameters of all gear drives. Most standards recommend that its value be higher than 1,15. But there are cases, when this cannot be achieved. Is the contact ratio factor below 1, high dynamic loads affect the teeth. There are not recommendations for such cases neither in the standard DIN 3990 nor in the standard ISO 3663 [10]. The low contact ratio can be taken into account by the factor K_A, but it certainly has its limitations. Our gearbox is a simple problem in this world because of its short lifespan, and because the contact ratio factor is only for a short time below 1, but managing low contact ratio is a major problem in many applications.

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