PNEUMATIC WRENCH GEARBOX DEVELOPMENT FOR TIGHTENING TORQUE OF 6000 NM

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Abstract
Pneumatic wrenches are a common tool in many industries. With increasing demands on their tightening torque, increases their size and price. For the application of a pneumatic actuator is also required a large gear ratio to achieve large tightening torques. Based on the requirements of company KOEXPRO Ostrava a.s., a project was created to develop a modular gearbox for pneumatic wrench. In combination with the already used NKH torque multipliers, it could be possible, shifting several gearboxes in a row, achieve a tightening torque of 6000 Nm. At the same time, such pneumatic screwdriver should not weigh more than 20 kg. Only in such case, it would be a product, which would overcome his competitors. This article describes the development of this modular gearbox.

Key words: *pneumatic wrench, gearing, modularity, planetary gearbox.*

INTRODUCTION
Pneumatic wrenches are a common used tool, when you need to develop a high tightening torque. They are simple, easy to use and can be uses in explosive environment. But when you need high tightening torques, they become big, heavy and not easy to handle. The company KOEXPRO Ostrava had only these big and heavy wrenches. The goal was to develop a pneumatic wrench, which would have tightening torque 6000 Nm a weigh no more than 20kg.

Fig. 1 Similar tool from the company PD Profi

The only solution for such extreme claims was to use the torque multiplier MKH and a lamellar pneumotor, which is light and has enough power. The only problem with this type of motor is his high rpm. The solution for this problem was to use a series of planetary gearboxes, which can develop the gear ratio. The minimization of weight of planetary gearboxes is described if the article of Predki W., Jarchow F., Lamparski Ch. (2001). Based on this article it was chosen the classical type A, given its simplicity and effectiveness. The aim of was to design such planetary gearboxes and this articles describes this development.

MATERIALS AND METHODS
The requirement for the design was to use the already produced torque multiplayer NKH 65. This multiplayer can be use up to 6200 Nm, weight's 9 kg and has a gear ratio of i_{NKH}=21. That means that for the planetary gearboxes and lamellar engine was just 11 kg, to achieve the require weight of 20 kg.

The decision was to use the lamellar pneumotor for the company Deprag. Pneumotors form this company have 5250 rpm. Because such wrench is used just for tightening bolts/nuts (not for screwing), After a consultation with the company KOEXPRO, it was decided that the output revolutions would be 1 per minute. That means that the overall gear ratio i_c would be 5250 and therefore than ten overall required gear ratio of planetary gearboxes i_{pc} would be 250.

Fig. 2 Currently used torque multiplayer NKH 65

The best solution was to use 3 gearboxes. For cost reduction, all gearboxes would be the same. That means that it is necessary that only the last gear stage forcibly go out correctly. That means that each planetary gearbox needs to have a gear ration i_p=6,3. The gearbox would be with braked crown wheel, with 3 satellites and the numbers of teeth are

Tab. 1 Number of gear tooth

Gear wheel	Number of teeth
Central	15
Satellite	33
Crown	-81

The real gear ration of this gearbox is than i_p=6,4. For cost reduction it was decided to use standard gear profile and to use module m=1 mm, that the weight and dimension are low as possible. The resulting gearing parameter ware:

Tab. 2 Geometrical dimension of gears

Material	Central	Satellite	Crown
Number of teeth z (-)	15	33	-81
Normal module m (mm)		1	
Correction x (-)	0.328541	- 0.328541	0.328541
Working axial distance a_w (mm)		24 (-24)	
Pitch diameter d (mm)	15	33	-81
Head diameter d_a (mm)	17,6	34,3	-78,7
Heel diameter d_f (mm)	13,157	29,843	-82,8
Gearing width b(mm)		50	
Contact ratio coefficient ε_α (-]	1,487		1,875

For strength calculation it was of course necessary to know the output power of the lamellar engine. The suggestion was that the hole pneumatic wrench would have an overall efficiency of η_c=0,58 (efficiency of torque multiplayer η_{NKH}=0,8, planetary gearbox η_p=0,9). The wrench is designed so it stops when it achieves the required torque (the required torque is set up by changing the air pressure). That means that the lamellar engine does not have to be designed on the nominal torque, but it is enough that is designed on the starting (maximal) torque. That means, that for the tightening torque 6200 Nm, the engine has to have 1,95 Nm. To this condition the company Deprag has the lamellar engine

63X-001F07, with nominal output power P_M=900W, nominal speed n_M=5200 rpm, nominal torque M_{kn}=1,6 Nm and maximum torque M_{km}=2,4 Nm.

It is of course clear that the maximal torque is not always required. For the stress calculation The suggestion was that the maximum load is in 30% of time, in 40% of time the gearing is loaded with 85%, and 30% of time is the gearing loaded with 70%. The load is of course not equal on all satellites, so the load was increased by 20%. The final load between central-satellite and satellite-crown wheel ware:

Tab. 3 Calculated loads on each gearing

No.	Part of time (-)	central-satellite (N)	satellite-crown (N)
1.	30%	4240	4240
2.	40%	3600	3600
3.	30%	2970	2970

The material properties ware one of the biggest problems. Of course the best solution ware to have tooth with hardened tooth sides, but this would increase the final price. The decision was made to use nitrided gearing, which does not require a subsequent processing. The material properties ware

Tab. 4 Used materials for gear wheels

Gear wheel	Material	Processing
Central	15 230.6	Nitrided
Satellite	15 230.6	Nitrided
Crown	12 061.6	Refined

The question was, if central gearwheel and satellite should be on all stages made out of the same material, or to use on first two stages a lower grade material. With another material the wrench would be cheaper. On the other side this would negative affect the modularity and interchangeability of parts and increase the risk of compliment. In the end the indirect costs by using a cheaper material, would increase the final price, that it is no difference if you use a more expensive material on all gearboxes, or you give on some stages a cheaper material.

RESULTS AND DISCUSSION

The strength calculation was is always the last step in gearing design. The calculation was performed according to ČSN 01 4686. The problematic part of the calculation was the contact stress control for the gearing central/satellite. This problem was described by Predki W., Jarchow F., Lamparski Ch. (2001) and Savage M., Rubadeux K. L., Coe H. H. (1998) in their articles. The safety coefficient s_H came out just over 1. Such result is not ideal, but the gearing was calculated for the worst case (reversing operation, shocks ...). The load is also the maximal assumed and in normal condition the wrench will be not used so often for torques around 6000 Nm. Such torque is for example the maximal tightening torque for screws M48 with grade 8.8., and they are not so common.

Tab. 3 Calculated safety coefficients for each gearing

	central-satellite		satellite-crown	
Safety coefficient	central	satellite	satellite	crown
Bending safety coefficient s_F	2,286	2,156	1,731	1,609
Contact stress safety coefficient s_H	1,012	1,042	2,195	1,770

For satellite/crown gearing ware the safety confidents for bending s_F and contact stress s_H almost equal, between 1,6 and 2,2.

To minimize the weight and maximize, it came into consideration using High Contact Ratio (HCR) gearing. After the first calculations, there was an increase of bending and contact stress safety coefficients and a loss of weight (thanks to a smaller gearing width). However, the improvements ware not so radical, that they would justify the additional costs by using HCR gearing. In the following proposals the concentration was to increase the contact ration coefficient ε_α, that it would be greater than 2. This measure would ensure, that calculated load in the gearing would be lower, so the gearing would be than thinner. The idea was, that when the number of tooth increases the diameter of gearbox will also increase, but when the value 2 of contact ration coefficient ε_α is reached, the width will the degrease. The lower width would then compensate the bigger gearbox diameter and the overall gearbox weight would be then lower.

However the reality was different. To achieve that ε_α>2 was not so difficult, but lower gearing width didn't compensate the bigger gearbox diameter as expected. Taking into account the higher cost of gearing tools and small number of wrenches pieces that would be produced, it was not cost-effective to use HCR gearing.

CONCLUSIONS

To design a planetary gearbox for a pneumatic wrench seems at the first glance as a not a hard task. But when you thing that the wrench should be for 6000Nm, powered by a pneumatic engine with 5250 rpm and weigh no more than 20 kg, then it is much more difficult task.
It was possible to design a planetary gearbox with standard gearing, which can withstand the expected loads. The calculated expected weight of each planetary gearbox is 3,45 kg, so the hole wrench should weigh 19,35, which is under the required weight of 20 kg. The problem with such proposal is to exactly specify the expected loads. If the gearbox would be calculated on full load for unlimited life the safety coefficients would then not come out. Also if the gearwheel should not be hardened, it is needed to use a high grade steel with corresponding surface finish.

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