Annals of DAAAM for 2010 & Proceedings of the 21st International DAAAM Symposium, Volume 21, No. 1, ISSN 1726-9679
ISBN 978-3-901509-73-5, Editor B. Katalinic, Published by DAAAM International, Vienna, Austria, EU, 2010
Make Harmony Between Technology and Nature, and Your Mind will Fly Free as a Bird
Annals of DAAAM International

A NEW PRINCIPLE FOR DIFFERENTIAL GEARBOXES

MARIN, D[umitru]; HADAR, A[nton]; IONITA, S[tela]; GRIGORAS, S[tefan] & SZABO, A[dam]

Abstract: *This paper presents a new principle to reduce speeds in a wide range of powers and reduction gear ratio. The advantages of this differential gearbox are presented too: small weight and sizes, high reduction gear ratio, two output shafts. The differential gearbox presented in the paper can be used in various fields: bucket wheel excavator and belt conveyors in mining; cement mills in cement industry; various winches for cable cars etc.*
Key words: *differential gearbox, gear ratio, speed, shaft*

1. INTRODUCTION

Many industrial machines such as: belt conveyors and bucket wheel excavators in mining, cement mills, etc. work with low speeds, about tens r.p.m. In this case, we use different type gearboxes, which are interposed between electric motor and equipment (Filipoiu & Tudor, 2006):

a. cylindrical and bevel gearboxes which have 2 ÷ 7 steps to reduce; they send any power and any gear ratio but they have two disadvantages: large sizes and great weight. Example: for a power of about 50 kW and a reduction gear ratio 800 ÷ 1000, weight can reach about 12.000 kg.

b. worm gearboxes are simple both building and functionally but they are low powerfully and efficiently.

c. planetary gearboxes make high reduction gear ratio at average power. They are advantageous due to their small sizes, but they are complexes and expensive (Bostan et al., 1997). Example: for a power of about 10 kW and a reduction gear ratio 1000 ÷ 1200, weight can reach about 2.000 kg but gearbox has many parts (14 gears and 6 shafts).

Exposed differential gearbox cumulates the three existent gearboxes types advantages but it faces less disadvantages. So, differential gearbox can send any power regarding those used everywhere at the necessary speed of the industrial equipment.

The differential gearbox advantage is that its sizes and weight are 30 ÷ 35% of conventional gearboxes.

2. DIFFERENTIAL GEARBOX DESCRIPTION AND OPERATION

A bevel pinion 1, gearings the set bevel gears 2, fixed by cylindrical gears 3 and 4. Bevel gears 2 have the same teeth number while gears 3 and 4 have one unity difference (see table 1). Both gear pairs are set on shaft 9. Cylindrical gears 5 and 6 also differ by a tooth while bevel gears 7 have the same number of teeth.

Satellite bevel pinions 8 are set on the shaft 11 which is fixed with the shaft 10. Cylindrical gear 5 ends in an output shaft of medium speed (70,5 rpm). Shaft 10 speed - the second output shaft - is the result of cylindrical gears 5 and 6 speed difference fixed with bevel gears 7.

Bevel gears 7 and satellite pinions 8 don't take part at reduction gear ratio; they are a sum mum device which provides algebraically difference of gears 6 and 5 (see equations 1 and 2).

No	Name	Module	Teeth	Speed
1	bevel pinion	m=4	z_1=13	n=1000 rpm
2	bevel gear	m=4	z_2=52	n=250 rpm
3	cylindrical gear	m=5	z_3=22	n=250 rpm
4	cylindrical gear	m=5	z_4=23	n=250 rpm
5	cylindrical gear	m=5	z_5=78	n=70,5 rpm
6	cylindrical gear	m=5	z_6=77	n=74,6 rpm
7	bevel gears	m=8	z_7=35	n=70,5 rpm n=74,6 rpm
8	bevel pinion	m=8	z_8=13	n=141 rpm

Tab. 1. Values for main parts of differential gearbox

$$n_6 - n_5 = 10^3 \cdot \left(\frac{z_1}{z_2} \cdot \frac{z_4}{z_6} - \frac{z_1}{z_2} \cdot \frac{z_3}{z_5} \right) \text{ rpm.} \tag{1}$$

$$n_6 - n_5 = 74,66 - 70,50 = 4,16 \text{ rpm.} \tag{2}$$

3. GEAR KINEMATIC RECKONINGS

3.1 For cylindrical gear 5 (average speed output shaft):
Number of teeth: z_1=13 ; z_2=52 ; z_3=22 ; z_5=78;
Input shaft speed: n_1 = 1000 rpm. In this case:

$$n_5 = \frac{z_1}{z_2} \cdot \frac{z_3}{z_5} \cdot 10^3 \text{ rpm} \tag{3}$$

$$n_5 = \frac{13}{52} \cdot \frac{22}{78} \cdot 10^3 \cong 70,5 \text{ rpm} \tag{4}$$

3.2 For shaft 11 (low speed output shaft):
Number of teeth: z_1=13; z_2= 52; z_3=22; z_4=23 ; z_5=78 ; z_6=77 ;
Input shaft speed: $n_1=10^3$ rpm. In this case:

$$n_{10} = \frac{z_1}{z_2} \cdot \left(\frac{z_4}{z_6} - \frac{z_3}{z_5} \right) \cdot 10^3 \text{ rpm} \tag{5}$$

$$n_{10} = \frac{z_1}{z_2} \cdot \left(\frac{z_3 + 1}{z_5 - 1} - \frac{z_3}{z_5} \right) \cdot 10^3 = \frac{z_5 + z_3}{z_5 \cdot z_6} \cdot 10^3 \text{ rpm} \tag{6}$$

$$n_{10} = \frac{13}{52} \cdot \frac{100}{78 \cdot 77} \cdot 10^3 \cong 4,16 \text{ rpm} \tag{7}$$

Note that the reduction gear ratio is primarily determined by the ratio between cylindrical gears teeth amount (z_3 + z_5) =(z_4 + z_6) and great cylindrical gears teeth product ($z_5 \cdot z_6$).

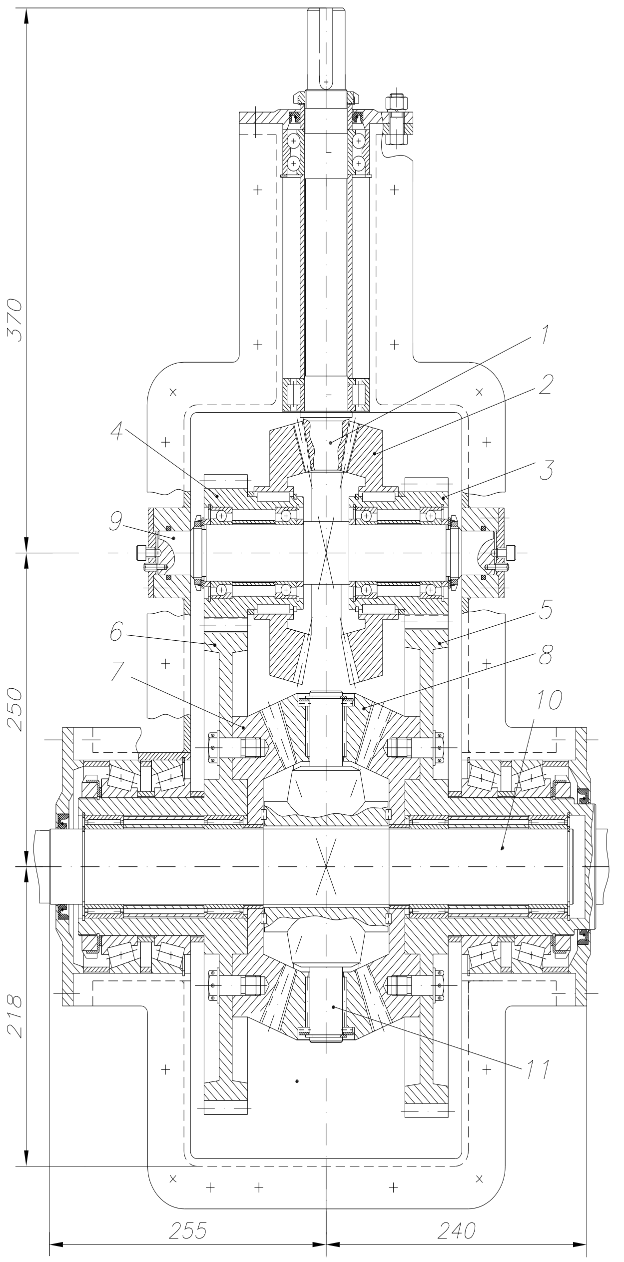

Fig. 1. Cross section on differential gearbox

If difference between teeth number of cylindrical gears would be two units then:

$$n_{10} = \frac{z_1}{z_2} \cdot \left(\frac{z_3 + 2}{z_5 - 2} - \frac{z_3}{z_5} \right) \cdot 10^3 \text{ rpm} \tag{8}$$

$$n_{10} = 2 \cdot \frac{z_1}{z_2} \cdot \left(\frac{z_5 + z_3}{z_5 \cdot z_6} \right) \cdot 10^3 \text{ rpm} \tag{9}$$

$$n_{10} = 2 \cdot \frac{13}{52} \cdot \frac{100}{77 \cdot 79} \cdot 10^3 \cong 8{,}22 \text{ rpm} \tag{10}$$

Note that the output shaft speed almost doubles over the case when cylindrical gears differ with one tooth.

If difference between teeth number of cylindrical gears would be three units then:

$$n_{10} = 2 \cdot \frac{13}{52} \cdot \frac{100}{77 \cdot 80} \cdot 10^3 \cong 12{,}15 \text{ rpm} \tag{11}$$

Note that the output shaft speed almost triples over the case when cylindrical gears differ with one tooth.

We have designed differential gearbox adopting values in table 1 (Constantinescu et al. 2007). Differential gearbox is represented in detail in figure 1 where we observe all parts and assembly (Marin, 2009).

In figure 1 are positioned only major component parts necessary to describe the differential gearbox operation presented in paragraph 2.

There were no major problems in design of differential gearbox or to achieve the experimental model.

Cooling problems may occur because of small sizes differential gearbox (because of the small amount of oil).

Thermal reckonings show that there is no need for forced cooling of differential gearbox (Grigoras & Stirbu, 2000*)*.

Differential gearbox will be built by NEPTUN Company in Campina, which is the most important producer of mechanical transmissions in Romania.

The main technical and constructive characteristics of the designed differential gearbox are presented in table 2.

No	Characteristics	Values
1	Rated power	10 kw
2	Reduction gear ratio	240
3	Rated speed of input shaft	1000 rpm
4	Maximum torque of output shaft	23200 Nm
5	Operating position	Horizontal ±5˚
6	Sizes: L x lx h	850 x 500 x 500 mm
7	Net weight	About 240 Kg

Tab. 2.Technical and constructive characteristics of gearbox

4. CONCLUSION

The presented differential gearbox, can successfully replace any of the classic gearbox: cylindrical or bevel gearbox, worm gearbox or planetary gearbox. They can send great power and high reduction gear ratio;

The great advantages of differential gearbox are: small sizes and weight (about 35% of those of conventional gearboxes);

Differential gearbox has two output shafts with different speeds, so two machines can be acted simultaneously. The low speed can be driven belt conveyors, cement mills, bucket wheel excavators etc;

Differential gearbox is reversible so it can function as a multiplier. In this case it can be used to convert wind power to low speed paddle shaft to synchronous speed drive electric generator.

Future research will follow choice of best materials and manufacturing technology for differential gearbox components.

5. REFERENCES

Bostan, I; Dulgheru,V. & Grigoras, St.(1997). *Sun-and-planet gears, processional and harmonics* , Ed. Tehnica, ISBN 973-31-1069-8,Bucharest

Constantinescu, I.N.,Radu, E.C. & Hadar, A. (2007). *Principles and methods of design in mechanical engineering-Computer aided design*, Ed. Printech, ISBN 978-973-718-786-4, Bucharest

Filipoiu, I., D. & Tudor, A. (2006). *Design of mechanical transmission*, Ed. BREN, ISBN 973-8143-26-8, Bucuresti.

Grigoras, St.; Stirbu, Cr. (2000*). Fundamentals of machine parts design*, Editura TEHNICA INFO,ISBN 9975-910-93-9, Chisinau

Marin,D.(2009) *Industrial Design - Shape Design*,Ed. BREN, ISBN 978-973-648-875-7, Bucharest