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THREE STEP MULTIPLIERS GEAR RATIOS FOR WIND TURBINES FROM MINIMUM VOLUME CONDITION

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ABSTRACT
This paper presents possible solutions of speed multipliers for wind turbines and how the transmission ratio must be divided on the three gear steps, considering the criterion of minimum volume of gears, for each of the possible solutions. Many studies have been developed in order to optimize the choose of the gear ratios of the consisting gears of a transmission, starting with different criteria, like: minimal summed centre distances, minimal volume of gears, minimal length, minimal width, minimal weight, minimal area of the frontal section of the transmission.
Keywords: optimal gear ratios, software, gear volume.

1. INTRODUCTION

The sustainable energy sources offer an inexhaustible energy potential and are available immediately. The utilization of sustainable resources will lead to a higher acceptance of renewable and to the spread of their use worldwide. The speed multiplier and the bearings are the most important parts of the wind turbines transmission. In comparison with other kinds of gear transmissions, the speed multiplier of the wind turbines has some main characteristics: the input torque is bigger than the output torque, the input rotation is smaller than the output rotation,

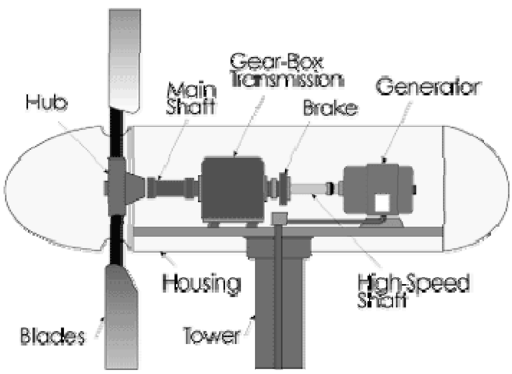

Figure 1. Schematic wind turbine components

the weight and the overall dimensions must be as reduced as possible [2,3].

There are few possible constructive solutions, from which here are represented in schematic form 4 of them [1]. In figure 2.a is represented a three stage external gears multiplier, the maximum value for the global multiplier ratio in this case is in a range between iM_{max}=100 … 150, 2.b represents a three stage multiplier the first two are external gears and the last is an internal gear, in this case the maximum value for the global multiplier ratio is in a range between iM_{max}=200 … 250. For both this constructive solutions 8 bearings are necessary, for all axes to be well sustained.

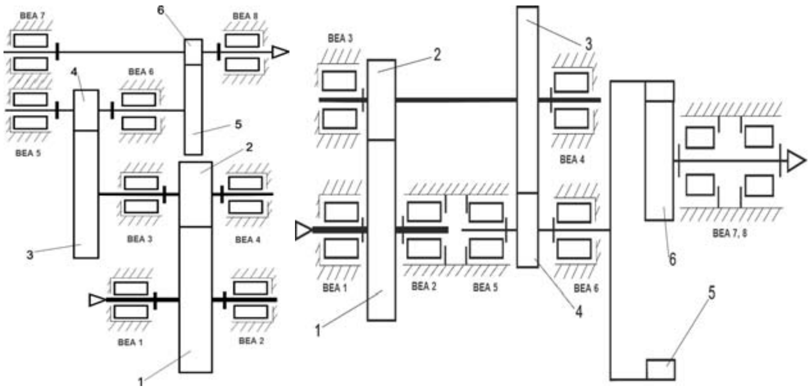

Figure 2. First type of solutions where the first step is an external gear

Other types of possible solutions are the ones represented in figure 3, the first step of the multiplier is an internal gear followed by two external gears, the maximum value for the global multiplier ratio in this case is iM_{max}=200 … 250, or in the other case an internal and an external gear where the maximum value for the global multiplier ratio is iM_{max}=280 … 330,.

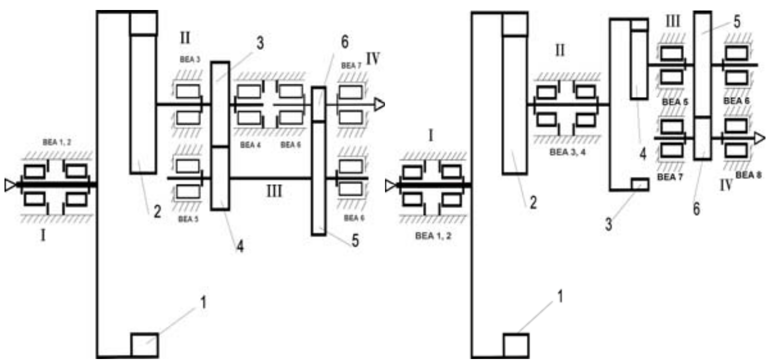

Figure 3. Constructive solutions with an internal gear in the first step

2. SOFTWARE

Software is developed and the results are presented as diagrams. This paper is imposing the criterion of minimal volume of gears, for an imposed global multiplying ratio(iM), the calculus is developed in the following steps set of the multiplying ratio in step I, (iMI) in the range of possible values (from iMmin to iMmax); Calculus of the multiplying ratio in step II and III; Gear volume (V) calculus; The optimal gear ratios (iMI, iMII and iMIII) are the ones for which the minimal gear volume (V) is obtained. Calculus is repeated for imposed global multiplier ratio (iM) in the range of possible values (from iMmin to iMmax), determining the optimal gear ratios, for each of these values.

The software developed in order to optimize a transmission with several gears used in the multiplier of a small or medium wind turbine. The process of optimization is mainly based on choosing the right gear ratio for each of the consisting steps of the transmission. Based on the logical flow from fig. 2, software was developed for determining optimal gear ratio considering the criterion of minimal volume of gears. This paper is dealing with choosing the optimized gear ratios of the consisting gears of three steps speed multiplier, in different constructive solutions.

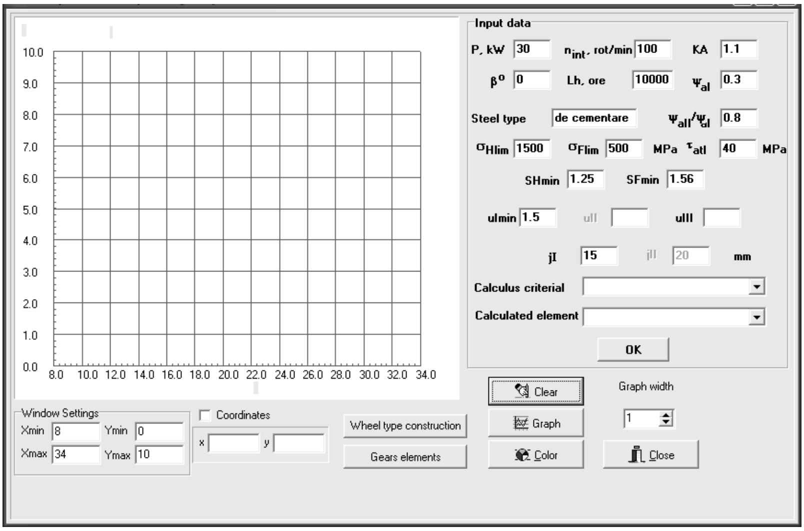

Figure 4. Results software interface

In order to choose the best transmission knowing the type of generator used and the power that the turbine can produce, it is necessary to know the value of the multiplier ratio iM, defined as the ratio of input angular speed to output angular speed:

$$i_M = \frac{\omega_{out}}{\omega_{inp}} \qquad \text{... (1)}$$

According to the wheel tooth number the multiplier ratio for a cylindrical gear with fixed axles can be determined with the following relation

$$i_M = \frac{z_1}{z_2} \qquad \text{... (2)}$$

where z_1 represents the driving wheel tooth number, and z_2 – the driven wheel tooth number.
For a three stage speed multiplier the transmition ratio can be determined by multiplying all three stage ratios:

$$i_M = i_{MI}\, i_{MII}\, i_{MIII} = \frac{z_1}{z_2}\frac{z_3}{z_4}\frac{z_5}{z_6} \qquad \text{... (3)}$$

The main imposed restrictions for the consisting cylindrical gearings are [4]:

- Avoiding the teeth profile interference;
- Achieving a minimum transverse contact ratio of $\varepsilon_{\alpha min}$=1.3;
- Avoiding the sharpening of the gearing wheels teeth $s_{a1} \geq s_{amin}$ and $s_{a2} \geq s_{amin}$;

- Choosing the minimal normal module according to the applied treatment. (m_{nmin} =1.5 mm, for case hardening, respectively m_{nmin} = 2.0 mm, for cementation or nitrating);
- The actual stresses for the two main stress types should not be bigger than the corresponding permissible stresses, $\sigma_H \leq \sigma_{HP}$; $\sigma_{F1} \leq \sigma_{FP1}$, $\sigma_{F2} \leq \sigma_{FP2}$;
- Tolerance of actual transmission ratio relative to imposed transmission ratio is ±3%;
- The gear ratio must be in the range of (1.5…8) for external gears and in the range of (2.5…10) for internal gears.

uI	awI	z1I	z2I	mnI	b2I	epsalfaI	uII	awII	z1II	z2II	mnII	b2II	epsalfaII	uIII	awIII	z1III	z2III	mnIII	b2III	epsalfaIII
5.28947	288.588627	38	203	3.5	87	1.96	3.41667	163 82	24	82	3	38.3442249	1.65	1.52381	119.2190634	21	32	4.5	28.92715801	1.52
5.28947	288.588627	38	203	3.5	87	1.96	2.80000	147.338050	25	71	3	34.8584652	1.62	1.56000	117.197085	25	39	3.5	27.3744874	1.51
5.34211	290.640404	38	204	3.5	88	1.95	3.47619	169.698279	21	73	3.5	40.0298026	1.54	1.54545	117.302589	22	34	4	27.30812111	1.50
5.39474	291.964417	38	205	3.5	88	1.96	2.84000	147.5782421	25	71	3	34.9111128	1.61	1.54545	117.3179734	22	34	4	27.2207688	1.52
5.48649	292.645945	37	204	3.5	88	1.94	2.96000	151.541601	25	73	3	35.48925924	1.64	1.52000	115.6285524	25	38	3.5	27.06368761	1.47
5.63889	295.522408	36	205	3.5	89	1.94	3.08333	152.401111	24	75	3	35.997349	1.60	1.50000	115.1548381	22	33	4	26.84793141	1.49
5.63889	295.522408	36	205	3.5	89	1.94	3.56522	162.976048	23	83	3	38.601254	1.58	1.52000	115.617180	25	38	3.5	26.8384889	1.54
5.80488	296.1911511	41	239	3	89	1.96	3.60870	163.504750	23	83	3	38.6448341	1.58	1.50000	114.829033	20	31	4.5	27.8831189	1.50
5.78049	295.4872897	41	238	3	89	1.97	3.64000	164.216833	25	91	2.75	38.68861501	1.62	1.52381	116.954469	21	32	4.5	29.0054060	1.51
5.80488	296.1911511	41	239	3	89	1.96	3.00000	148.459828	24	72	3	35.1030676	1.55	1.52381	116.795899	21	32	4.5	29.00467374	1.51
5.80488	296.1911511	41	239	3	89	1.96	3.68000	166.730267	25	92	2.75	38.82112361	1.66	1.52632	117.236839	19	29	5	28.980184	1.52
5.90000	296.340556	40	238	3	89	1.94	3.04167	150.115193	24	73	3	35.1599456	1.63	1.52381	117.4343484	21	32	4.5	28.8789971	1.53
5.90000	296.340556	40	238	3	89	1.94	3.16667	153.812392	24	75	3	35.7937215	1.65	1.54545	114.005170	22	34	4	27.3885775	1.48
5.80488	296.1911511	41	239	3	89	1.96	3.26087	152.696780	23	76	3	36.1617179	1.56	1.58333	114.030860	24	38	3.5	26.51361754	1.50
5.97500	297.767494	40	239	3	90	1.95	3.30435	154.498121	23	77	3	36.32581634	1.60	1.54167	112.426176	24	37	3.5	26.27484527	1.44
5.90000	296.340556	40	238	3	89	1.94	3.83333	165.347263	24	93	2.75	39.09106767	1.59	1.56522	119.357878	23	35	4	27.9177475	1.60
6.02500	299.393910	40	240	3	90	1.94	3.87500	165.824254	24	93	2.75	39.1366586	1.58	1.54167	112.353005	24	37	3.5	26.2004308	1.46

Figure 5. Table results – Gears elements

In the calculus is taken into consideration the fact that the internal wheel is ring shape.

3. CONCLUSIONS

Calculations have been developed for a speed multiplier with the following inputs: input power $P=30\ kW$ input rotational speed $n_{inp}=100\ rot/min$, application factor $K_A=1.1$, helix angle for both gears $\beta=0°$, total imposed running time $Lh=10000\ h$, width coefficient $\psi_{aI}=0.3$, width coefficient ratio $\psi_{aII}/\psi_{aI}=0.8$, $u_{min}=1.5$ for external gears and $u_{min}=2.5$ for internal gears, permissible torsion stress for shaft pre-dimensioning $\tau_{atI}=40\ MPa$, clearance $j_I=15$, $j_{II}=20$ gears are made of cementation steel with $\sigma_{H\,lim}=1500\ MPa$, , $S_{H\,min}=1.25$, $\sigma_{F\,lim}=500\ MPa$, , $S_{F\,min}=1.56$.

The software developed allows the user to have more possibilities from which to choose the one that most fit the application. An important feature of this software is that way that the results are displayed. All the results that came out after all calculations of the software are then inserted in a table, from which the user can study all the gears dimensions and decide which one fits best on the application requirement. After the gears calculations, the bearings are also choose depending on the values of the forces that the gears are developing and that will load the bearings. The life of the wind turbine is guaranteed for 20 years, the gears where supposed to operate for 10000 hours in normal conditions and without maintenance, then the life of the bearings must arrive at the same amount of time, if the life of the bearings are miscalculated the life of the wind turbine can be influenced.

4. REFERENCES

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