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DESIGN OF A GEAR AIR ENGINE WHIT USE OF „HCR" GEARING

NÁVRH ZUBOVÉHO PNEUMOTORU S VYUŽITÍM „HCR" OZUBENÍ

Abstract

In the design of a gear air engine is the size of the tooth space fundamental. Most current calculations, however, cannot, especially for non-standard "HCR" gearing, this problem satisfactorily describe.Therefore, it is preferable to use for this type of air engine „HCR" gearing, which maintaining the desired performance, while reduce the overall dimensions of the air engine. This article shows the possibility of calculating the exact size of the tooth space using numerical methods and shows the advantages of a gear air engine whit „HCR" gearing, compared to a gear air engine whit standard gearing.

Abstrakt

Při návrhu zubového pneumotoru je zásadní velikost zubové mezery. Většina současných výpočtů, však nedokáže, hlavně pro nestandartní „HCR" ozubení, tento problém uspokojivě popsat. Z toho důvodu nelze plně využít výhod „HCR" ozubení - při zachování požadovaného výkonu, dojde ke zmenšení celkových rozměrů pneumotoru. Tento článek ukazuje možnosti přesného výpočtu velikosti zubové mezery za pomocí numerických metod a ukazuje výhody pneumotoru s „HCR" ozubením v porovnání s pneumotorem se standartním ozubením.

1 INTRODUCTION

Development of the air engine is carried out in cooperation with KOEXPRO Ostrava a.s. and it should serves to drive a mine winch (the device is intended for the transport and movement of materials, machines and their parts in the mines on horizontal lines). Reversing of the motor will be done by changing the direction of the air flow in the pipeline.

2 PROPOSAL OF THE MOTOR DIMENSIONS

Input values, which I based on, were given by the requirements of company KOEXPRO:

- Minimum power P = 4 kW
- Air pressure p = 0.3 ÷ 0.6 MPa
- Working revs n = 1900 min-1

In the initial design of the air engine, I used values from MPZ4 air engine from KOEXPRO Company. The gearing parameters of this engine were:

- Module m = 9 mm
- Number of teeth = 12

For the design I chose, however, the unit addendum h_a *= 1.3, the proportional head clearance c*= 0.15. For a better meshing characteristics and also easier manufacturing, I decided to use on each shaft 2 symmetrically mounted wheels whit helical gearing, which creates a double helical gearing. I made the proposal for a half gearing and half the desired output power.

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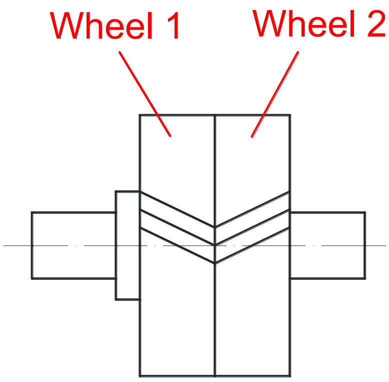

Fig. 1 Mounting of the gear wheels.

From the input values, I calculated the required width of the teeth. This is also an important factor that influences the size of the total contact ratio coefficient ε_χ. It was therefore necessary to design the gear so that the total contact ratio coefficient ε_χ was a whole number (with a maximum deviation of ± 2%) and at the same time that the engine had the required minimum output power.

For this reason, I wrote a computer program which for the specified engine and his gear parameters pre-calculated the minimum required width of the gear (according to [1]). Then, after selecting the real gear width, it calculates the real engine output power.

3 CALCULATION OF GEOMETRIC VOLUME

The basis for the calculation of the engine parameters is the calculation of the geometric volume. At the beginning it was necessary to determine the work space in which does the air the work. I assume that this space is between the tip circle d_a and the calculation circle d_v. Calculation circle d_v is centered in the center of wheel 1 and it is also tangent to the tip circle of wheel 2.

$$d_v = 2 \cdot \left(a_w - \frac{d_a}{2} \right) = 2 \cdot a_w - d_a \tag{1}$$

Height of the work area h is then

$$h = \frac{d_a - d_v}{2} = a_w - d_a \tag{2}$$

The geometric volume Vg is then given by the difference between the annulus surface (between the diameters d_a and d_v) and the total transverse teeth flank surface S_c, multiplied by the width of gear b.

The total transverse teeth flank surface is calculated from the flank surface of one tooth, multiplied by the number of teeth z. It is calculated with surface, which height is the height of the workspace h (see fig. 2.).

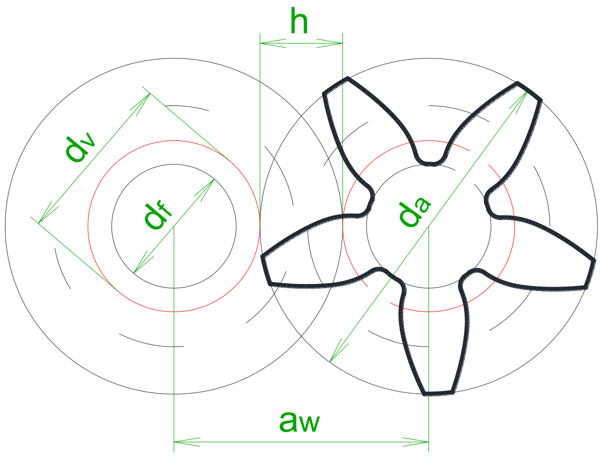

Fig. 2 Chart of the work space.

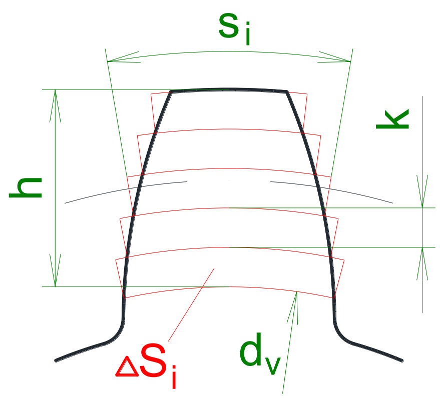

Fig. 3 Calculation of the tooth surface.

The flank surface of one tooth S, is then given by the sum of each ΔS_i, where n is the number of steps in the calculation.

$$S = \sum_{1}^{n} \Delta S_i \tag{3}$$

Surface ΔS_i is then calculated so, that it is calculated the transverse tooth thickness on a given diameter, which is then multiplied by the step size k.

$$\Delta S_i = s_i \cdot k = s_i \cdot \frac{h}{n} \tag{4}$$

The calculation of the tooth thickness on a arbitrary diameter is based on [2] p. 73, but I modified it for helical gear. The calculation of the tooth thickness on a arbitrary diameter for helical gear is very complex, since on each diameter is needed to calculate the transverse pressure angle α_{ti} and the helix angle β_i. The influence of a numeric calculation inaccuracy, due to low demands on performance computing computer, are eliminated by a large number of steps n (n = 10000).

Calculation according to [3], can be used however only in case when the calculation diameter d_v is bigger as the base diameter db. But if $d_v < d_b$ then the calculation needs to be modified. In this case according to [3] begins the calculation on the base diameter d_b. Additional tooth surface S_d, between circles d_v and d_b, is then calculated by

$$S_d = s_b \cdot \frac{d_b - d_v}{2} \qquad (5)$$

Where s_b is the tooth thickness on the base circle. Total tooth surface S_{zc} is then given by

$$S_{zc} = S + S_d \qquad (8)$$

The total surface of the tooth space S_{zm} is given by the difference of the annulus surface and the total transverse flank surfaces of all teeth, minus half of the surface of head circles intersection S_p, which I assume that the entire space is filled teeth.

$$S_{zm} = 2 \cdot \left(\frac{\pi}{4}\left(d_a^2 - d_v^2\right) - z \cdot S_{zc} \right) - \frac{1}{2} S_p \qquad (7)$$

Here I assume that the entire space is filled teeth

$$S_p = \frac{d_a^2}{2} \cdot \arccos\left(\frac{d_a - h}{d_a} \right) - \left(d_a - h\right) \cdot \sqrt{\frac{d_a \cdot h}{2} - \frac{h^2}{4}} \quad (8)$$

The resulting geometric volume of the air engine V_g is then given by multiple of the surface S_{zm} and the engine width b.

$$V_g = S_{zm} \cdot b \qquad (9)$$

4 RESULTING PARAMETERS

I chose the width of one gear wheels b = 76mm (the width of hole double helical gear is B = 2·b = 152 mm), so that the engine had the required output power, which finally is P = 4.68 kW. Air consumption of the engine is then 4.52 m^3/min.

The old design had the output power P = 4.07 kW, by air consumption of the engine 4.07 m^3/min.

To other calculations, for example Kopáček [1], is the geometric volume (or rather power) of the engine about 5% bigger.

The resulting parameters of the gear wheel finally are (the width b is just width of one gear wheel):

Tab. 1 Gear dimensions.

Gear parameters			New design	Old design (MPZ4)
Number of teeth	z	[-]	10	12
Normal module	m_n	[mm]	7	9
Profile addendum factor	h^*_a	[-]	1.3	1
Profile tip clearance factor	c^*	[-]	0.15	0.25
Profile roof rounding radius factor	ρ^*	[-]	0.228	0.38
Helix angle at reference diameter	β	[°]	33	0
Normal flank angle	α_n	[°]	20	20
Working center distance	a_w	[mm]	88.5	113
Addendum modification coefficient	x	[-]	0.4094	0.32002
Centre distance factor	Δy	[-]	0.0997	0.08395
Root diameter	d_f	[mm]	66.6979	91.2603
Base diameter	d_b	[mm]	76.5659	101.4868
Beginning of involuce diameter	d_L	[mm]	76.5898	101.4934
Working diameter	d_w	[mm]	88.5000	113.0000
Reference diameter	d	[mm]	83.4654	108.0000
Tip diameter	d_a	[mm]	105.5000	130.2500
Common facewidth	b	[mm]	76	120
Specific sliding at root		[-]	-3.48391	-3.60107
Specific sliding at tip		[-]	0.77698	0.78266
Transverse contact ratio	ε_α	[-]	1.172	1.205
Overlap ratio	ε_β	[-]	1.882	0
Total contact ratio	ε_γ	[-]	3.054	1.205

5 BRAKE

To the requirements of company KOEXPRO included also the fact that the engine be equipped with a parking brake, released by the air pressure in the engine. However, on the market is no brake whit so low desired release air pressure and in the required dimensions, so I had to brake to design myself.

From the need to have smallest dimensions as possible, I decided to design a multi-plate brake, whit a friction plate combination steel-organic, from company Ortlinghaus [4], because of the high coefficient of friction and the opportunity to run dry.

6 CONCLUSION

The result is an air engine proposal, which dimensions and weight are due to use „HCR“ gearing, compared to other air engines, considerably smaller. But the use of two symmetrically mounted helical gears on one shaft (to create a double helical gear), creates in the end a more complicated engine design. This is given by the need to tie up both wheels together to prevent leakage of the compressed air. At the same time the wheels on one shaft must allow an axial displacement, to allow balance ion of inaccuracies in the gear.

Results show that the output power has increased by 15%, but the air consumption has increased just by 11%. Despite this output power increase, the total built-up volume decreased by approx. 22% compared the original engine MPZ4. At the same time, thanks to the use of double helical gearing, there was a significant improvement of the contact ratio properties. The total contact ratio coefficient $\varepsilon\chi$ increased from value 1.205 by MPZ4, to 3.054 by the new design. In result of this, the engine whit the new design is much quieter and produces less vibrations, compare to the MPZ4. Also has the new design, thanks to the use of “HCR” gearing, lower specific sliding in the gearing. As result of that, the gearing on the new design wears less out.

The disadvantage by use of a „HCR“ gearing is the fact, that there may be such a large dimension reduction of the engine, the other attached components, such the brake, are for this performance parameters and so small dimension normally do not produce and must be custom-made.

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