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SOME ASPECTS OF EMBODIMENT DESIGN

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Abstract: *In the paper some aspects of embodiment design starting from the principle solution are discussed. The compulsory use of standardized elements (and dimensions) or of integer numbers of elements (or only details of these, ex. teeth) represents a difficulty. Because the actual usual computers permit the use of repetitive complicated calculation, optimal solutions for different size determining problems of embodiment design can be made. Specific methods for optimized size determination of translation screw and of helical gears are presented.*

Key words: *optimized choose, standardized dimensions, translation screw, gear.*

1. Introduction

In domain of product development the embodiment design consists in developing of part design starting from the principle solution. This involves a large number of corrective steps, in which the analysis and the synthesis constantly alternate and reciprocally complement each other. This activity uses the different type of requirements (referring to spatial limits and material) to determine the shape, size and arrangement of components [5].

A difficulty consists in the compulsory use of standardized elements (and dimensions) or of integer numbers of elements (or only details of these, ex. gear teeth). Because the standardised dimensions are not of continuous type and the detail numbers are not positive real one, it is necessary to adequate the calculus in order to choose of these ones.

There is no fixed or imposed machine design procedure for the calculus of machine element of the machine is being designed a number of options that have to be considered.

Ourselves we imposed some methods or rules, but all these have an historic moment of true. When designing machine one cannot apply rigid rules to get the best design for the machine and in the same time to attend the lowest possible cost. The designer who develops the habit of following only a fixed algorithm of steps for designing mechanisms or machine elements cannot expect to design himself the best product. When the new product is to be developed the problems can begin even at design level stage and these can be solved only by having a flexible approach and considering various ways to design based on knowledge.

We consider that the machine design procedure is not a standard one and due to this assumption there are some common steps to be followed; These steps can be more complex and can be followed as per

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the requirements wherever are necessary.

The afferent calculation is possible today using usual computers.

In the paper aspects of several examples of adequate activity for the right choose of characteristics are discussed.

2. Choose of correct variant thread for nut screw translational mechanism

The translation screw or power screw is used to translate the turning motion into linear one in linear actuators, presses, jacks etc. Two principal forms of translation screw threads are used: trapezoidal and buttress. Because the loading force is generally greater in one direction as in the other, both the screw threads can be used.

In case of jacks or presses the screw body is compressed. For the calculus of the size, two minimal diameter values are calculated imposing the conditions [2]:

- from condition of load capacity assurance at compression and at combined stress (compression and torsion);
- from condition of buckling avoiding.

The maximum of these ones is kept.

Following requirements must be fulfilled:

a. **compulsory demands**:

- the limiting the number of thread turn in contact (z),
- the assuring of sufficient strength of thread (fulfil by metallic and standardized screw and nut) and
- the thread self locking [2], [3]

$$\alpha < \varphi' = \arctan\left(\frac{\mu}{\cos(\beta/2)}\right), \tag{1}$$

with ε ï coefficient of friction, ϐ ï thread angle, Ŭï the lead angle [2], [3]

$$\alpha = \frac{p}{\pi \cdot d_2}, \tag{2}$$

with p ï pitch, d_2 ï pitch diameter;

b. **objectives for optimal calculus** (called wishes by Pahl and Beitz [5])

- minimal torque, maximal efficiency and (by same performances)
- the trapezoidal thread is preferable because of profile symmetry.

Although theoretically the buttress thread assure a lower value of torque as the trapezoidal one, because of discrete dimensional standardised series, in a lot of cases the trapezoidal profile is a better choice.

The authors consider that a lot of dimensional variants of trapezoidal (after ISO 2904 [10]) and buttress thread (after DIN 513-2 [8]) having the root (minor) diameter greater as d_{3min} must be compared. These are analysed:

- for two (or more) materials of nut (as ex. bronze and cast iron) and;
- for two coefficient of friction (corresponding to poor and abundant lubrication),

In accomplish the calculus following elements are to be calculated:

- lead angle (Ŭ) of the screw helix and friction angle (ű ô) between the materials of screw and nut;
- number of thread turn in contact [2]

$$z = \frac{4 \cdot F}{\pi \cdot (d^2 - D_1^2) \cdot p_a} < 10, \tag{3}$$

with F ï loading force, D_1 ï minor diameter of internal thread, p_a ï admissible contact pressure;

screw torque [2], [3]

$$T = F \cdot \frac{d_2}{2} \cdot \tan(\alpha + \varphi'), \tag{4}$$

thread efficiency [2], [3]

$$\eta = \frac{\tan(\alpha)}{\tan(\alpha + \varphi')}. \tag{5}$$

In example presented in table 1 the mentioned elements are calculated for:

- loading force F = 20000 N,
- ε å 0.15 (poor lubrication) and
- ε å 0.08 (abundant lubrication),
- admissible pressure p_a å 12 MPa (steel / cast iron) and
- p_a å 20 MPa (steel / bronze),
- for each 5 variants of trapezoidal and

Table 1

Thread	d_3 [mm]	z (p_a=12 MPa)	z (p_a=20 MPa)	Ǔ[°]	ε=0.15			ε=0.08		
					ű'[°]	T [Nmm]	ɖ	ű'[°]	T [Nmm]	ɖ
Tr 24x3	20.5	15.719	9.431	2.43	8.827	44790	0.21	4.735	28280	0.34
Tr 25x3	21.5	15.05	9.03	2.327	8.827	46340	0.20	4.735	29110	0.33
Tr 25x5	19.5	9.431	5.659	4.046	8.827	51420	0.31	4.735	34750	0.46
Tr 26x3	22.5	14.436	8.661	2.232	8.827	47890	0.2	4.735	29940	0.32
Tr 26x5	20.5	9.03	5.418	3.874	8.827	52970	0.3	4.735	35580	0.45
S 26x3	20.794	9.928	5.957	2.43	8.827	45500	0.21	4.735	28670	0.338
S 28x3	22.794	9.157	5.494	2.327	8.827	48500	0.197	4.735	30270	0.328
S 28x5	19.732	5.834	3.5	4.046	8.827	52860	0.301	4.735	35530	0.458
S 30x3	24.794	8.497	5.098	2.232	8.827	51500	0.185	4.735	31870	0.319
S 30x6	19.586	4.623	2.774	3.874	8.827	58050	0.329	4.735	39770	0.447

buttress thread having d_3 greater as 19.34 mm.

Comparing the results following observations can be made:

- some variants have the number of thread turn in contact greater as 10 for p_a å 12 MPa;
- all variants respect the self locking requirement in both lubrication conditions;
- all torques are great for manual action and as a result the efficiency (ɖ) becomes unimportant;
- the buttress thread variants assure better results for the number of thread turn in contact (z), but a good result gives the choice of the thread Tr 26x5. This one has the major diameter equal to the minimal one of buttress threads;
- the torque values differ not significantly for both thread types (due to the discrete distribution of dimensional characteristics of standardized threads);
- for this example the buttress thread offers not significantly advantages in order to compensate the disadvantage of profile asymmetry (that may cause manufacture and assembly errors);
- following variants can be kept for ulterior analysis:

1. Tr 24x3 with nut made from bronze material, eventually with abundant lubrication (not compulsory);
2. Tr 26x5 with nut made from cast iron material and abundant lubrication.

3. Size determination of a gear

Afterwards the case of size determination of a helical gear is discussed. The group of standards ISO 6336 Parts 1é 5 ([11]é [15]) resolve only the problem of calculation of load capacity of already size-determined spur and helical gears.

The first sub step of gear embodiment is the determination of its size-characteristics. This consists in:

- calculation of a minimal centre distance (a_{min}) or of a minimal pinion diameter $d_{1\ min}$);
- choose of suitable values of these ones;
- choose of face width (b);
- calculation of minimal module;
- choose of a convenient value of module (m_n);
- determination of profile shift

coefficients (x_1, x_2);

- calculation of geometrical characteristics of gear
- verification of loading capacity using ISO 6333 ([11]÷[15]) or other adequate method.

In Romania the most used size-determining methods for gears are the ones of Niemann and Winter [4], Henriot [1] and Rădulescu [6], [17].

The method Niemann - Winter [4] uses an empirical factor (K^*) - dependently of operating domain of gear – in order to calculate the minimal pinion diameter (d_{1min}). In same way, using another empirical factor (U), the minimal module ($m_{n\ min}$) is determined.

Therefore the method is applicable only for some domains, for which these factors (K^* and U) are given in the tables from [4].

Using the Henriot method [1] the minimal pinion diameter (d_{1min}) and the minimal module ($m_{n\ min}$) are determined. Here the operating domain of gear is considered by the service (application) factor (K_A). Its assessment is a difficulty. Henriot gives the same values for K_A as the ones from ISO 6336-6 [15] and, for more accuracy, recommends the list from AGMA.

It is to observe that the indications from ISO 6336-6 [15] are richer as the ones from [1]. In ANSI/AGMA 6110-F97 [7] is given a detailed list containing values of factor K_A (denominated here as "service factor") depending on application domain. These values are to be corrected by converting factors (also given here) in order to consider the engine type influence.

The main application domain of this method (with its empirical factors and recommendations) is the one of aeronautical constructions.

The method Rădulescu ([6], [17]) uses the influence factors given in ISO 6333 ([11]…[15]) in order to obtain the minimal gear dimensions, i.e. centre distance (a_{min}) and module ($m_{n\ min}$). Initially mean values for those factors are given. If the result of load capacity calculation is unfavourable, new values of some influence factors (calculated – at load capacity assessment - and having great deviations from the means preliminarily adopted) are used to recalculate the minimal dimensions (a_{min} and $m_{n\ min}$).

The same problem as the one occurred at Henriot method [1] consists in adoption of an appropriate value of application factor (K_A) and the same observations are here valid.

If the used method gives the minimal pinion diameter (d_{1min}), the minimal centre distance can be approximated

$$a_{min} \approx \frac{d_{1\,min} \cdot (u+1)}{2}, \qquad (6)$$

with u - the gear ratio.

A greater value of centre distance (a) as the minimal one (a_{min}) is chosen. For great gears, for planetary ones, for car gears or other concrete cases only a rounded value is chosen. In case of dimensional series standardized values from ISO 3 [9] or other norm are adopted.

It is to observe that the mentioned methods ([1], [4], [6], [17]) give indications to determining the face width (b). In some special cases other literature recommendations are to be used.

Following, only the case of power gears is discussed. The kinematical ones (ex. the gearbox of a lathe) are excepted because their compulsory gear ratio value (u).

The size determination of helical gear consists in determination of:

- teeth numbers (z_1, z_2),
- module (m_n) and
- profile shift coefficients (x_1, x_2).

The methods Niemann-Winter [4] and Rădulescu [6] give domains depending on gear material and gear ratio (u) for choosing the pinion teeth number (z_1).

Following **compulsory requirements** must be fulfilled:

- z_1 must be chosen from recommended domain;
- z_1 and z_2 without common divisors;
- maximal relative deviation of gear ratio must fulfil the condition

$$\frac{|u - z_2 / z_1|}{u} \le 0.03\,; \tag{7}$$

- the sum of profile shift coefficients must be in the domain

$$0 \le x_1 + x_2 \le 1.2\,. \tag{8}$$

A wish (desired non-compulsory requirement) for increasing the bending strength of teeth is

$$x_1 + x_2 \to 1\,. \tag{9}$$

The authors propose following method:

a. generating of teeth numbers pairs (z_1, z_2) for whole recommended domain of z_1 respecting the requirement given in (7);

b. elimination of variants (z_1, z_2) having common divisors;

c. calculation of sum of profile shift coefficients (x_1 + x_2) after [4] [6] [17];

d. retaining of variants (z_1, z_2) which fulfil the requirement (8);

e. distribution of the sum of profile shift coefficients between the mating gears, using the method from ISO 21771 [14] or the one given in [4], [6] and [17].

For illustrating the proposed method this was applied in case of a gear having $a = 180$ mm, $u = 5$ and [13é 20] as recommended domain for chosen of z_1. The results are given in table 2.

In is to observe that:

- all variants from table 2 fulfil the requirements presented above;
- the second variant (z_1=17, z_2=83, m_n=3.5 mm) assure a better sum of profile shifts (conforming to desired requirement shown in expression (9)).

Table 2

z1	z2	m_n [mm]	x1+x2	|φu|/u [%]
15	73	4	0.329482	2.66
17	83	3.5	0.687092	2.35
17	84	3.5	0.151108	1.17
20	97	3	0.618826	3.

4. Conclusions

1. The actual usual computers permit the use of repetitive calculation for analyzing comparatively the possible solutions in order to find quickly optimal solutions for different size-determining problems of design.
2. Specific methods for optimized size-determination of translation screws and of helical gears are presented.
3. By transmissions with translation screws that are manually acted the limit active moment cannot be exceeded and the minimizing of resistant moment is a important choosing criterion. The efficiency becomes unimportant.
4. Because of discrete distribution of dimensional characteristics of standardized threads, in a lot of concrete cases the trapezoidal thread offers better performances as the buttress one.
5. By similar performances trapezoidal thread (having symmetrical profile) is preferable to the buttress one in order to avoid the possible manufacture and montage errors.
6. Useful values for application factor K_A can be found in ISO 6336-6 [15] and especially in In ANSI/AGMA 6110-F97 [7].
7. Because the teeth numbers must be integer, the gear ratio is difficultly obtained using prime numbers.

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