SOME CONSIDERATIONS FOR PRECISION AND FITS CHOICE IN BUILDING MACHINES

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In the paper are presented some aspects in regard to the precision choice, the fit's group and optimal way of fit, used in building machines.
The optimal designing of clearance, interference and transition fits in base of basic shaft using the „key" relation and the made table with numerical examples represents the base of establishing algorithms and programs for computing design.
Keywords: utilization key, monograms, abacus, adjustment.

Introduction

With a great importance in the technical, functional and economic point of view, the grade of tolerance of the piece manufacturing and the fits are established and chosen corresponding to the achievement possibilities, to the assembling and processing economical, to the functional parameters which are imposed by the operation and running conditions and to other factors.

The types of the initial data which are well known by the designers in the designing background, when needs to prescribe a fit are:
a) only the qualitative neither the quantitative fit aspect are known;
b) the quantitative fit aspect are known too and, in this case, is imposed the extreme edge of the assemblage characteristics (J_{max} and J_{min} ; S_{max} and S_{min}).

In the case of point a) situation, these steps are made:
- the fit system is chosen from the two possibilities: the fit (assembly) system with basic hole and the fit system with basic shaft. Anyway is recommended the fit system with basic hole. The fit system with basic shaft is chosen only for then assemblies which technological or functional conditions impose this system;
- is established the fit type (the fit character): clearance fit, transition fit (passing by or uncertain) or inference fit;
- the basic fit is chosen in function of this domain literature recommendations (books, papers, standards, instructions etc.), standard's settlements, designer experience, but even of the respective ensemble characteristics (the temperature domain, the volume and the types of the strains the speed, assemblage length, the number of the supports etc.). In the choice process of the grade of tolerance levels for the shaft and the fit is taken into consideration the principle "rougher possible, but with as small tolerance as necessary to running"[2];
- computing the deviations.

In the case of point b) situation when appears the problem of establishing of the optimal fit type, the grade of tolerance and the proper fit selections are not so simple.

The grade of tolerance levels (ISO standards), in the case of fits, are chosen in function of the following factors [1]:
– assemblage quality- which lay in the assembled parts property to be, n dimensions, closest to the nominal value mentioned in the diagram;
– the functional determinant level of the assemblage which is bigger than the assemblage tolerance characteristic is smaller vice versa.

The grade of tolerance level selection

The designer, in the grade of tolerance level selection process, follows these steps:
I. On the base of functional conditions, is imposed a variation of the assemblage characteristic (clearance or interference), proper to a certain assemblage, between two approximate limits, like an example:
J = (0,009...0,052) mm, where the clearance tolerance is:

$$T_j = J_{max} - J_{min} = 52 - 9 = 43 \mu m \tag{1}$$

From the general expression of the tolerance:

$$T = ITx = Cx \cdot i \tag{2}$$

is computed the number of the tolerance steps Cx_{aj}, for the clearance fit which is considerate, even for the nominal dimension, at 35 mm i.e. in the domain (30...40] mm with $\phi_m = 35\ mm$:

$$Cx_{aj} = \frac{T_j}{i} = \frac{T_j}{0{,}45\sqrt[3]{\phi_m} + 0{,}001\,\phi_m} = \tag{3}$$
$$= \frac{43}{1{,}5069} = 28{,}53 \cong 28$$

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for each twin parts of the fit : $\frac{Cx_{aj}}{2} = \frac{28}{2} = 14$.

From the table of the tolerance units number, in function of the IT characteristics (table1), results, for the analyzed fit example, class 6 (which has(C6=10) and class (C7=16)).

Since it was established the precision classes 7, respectively 6: $\phi N \frac{7}{6}$ (on the base of the fact that the shaft can be manufactured more refined at the some cost), results: $C6 + C7 = 10 + 16 \cong 28$.

These characteristics will satisfy the tolerance parts values which will ensure the clearance imposed by the functional conditions.

Table 1

Calităţi ITx	5	6	7	8	9	10	11	12	13	14	15	16
Numărul unităţilor de toleranţă Cx	7	10	16	25	40	64	100	160	250	400	640	1000

II. The second criterion is detailed presented in [1], [3], [4], [5], [6], [7].

The fit type selection for the clearance group

In table 2 [1], [5] is presented the condition which appears at the clearance fits in one of them "moments of life", beginning with assembling, continuing with working and ending with disassembling of the twin parts.

In the following paragraphs is about the selection of the fit type.

On the ground of the running, from table 2, the fits can have these behaviors:

A) Statically (F^*), groups II, III and IV, when the fit type "is selected" in this way that the assemblage needs dynamics only at installation. For maximal grade of tolerance maintaining will be selected the fit H/h. Can be selected even the transition fits H/js or H/j which are clearance fits actually. For no importance assemblages, to have easiness in assembling can be selected the fit from the beginning of the alphabet (H/g; H/e ...).

B) The fit has a dynamical working (M). In this case, the designer can meet two variants:

B.l. *He has the technical and functional parameters like the data input of the problem;*

B.2. *There are imposed the admissible extreme characteristics of the assemblage (J_{max} and J_{min}).*

In the case B1, t is computed the average clearance (J_{med}), i.e. functional optimum clearance, and after this is composed and used the table 3, on the ground of the basic hole, and from [1], (in [5] is presented the table composing and using on the ground f the basic shaft) can be computed:

1. the fit type;
2. the grade of tolerance level for each piece in part;
3. the admissible deviations from nominal dimension of the two twin parts;
4. the assemblage characteristics;
5. the fit correction at the manufacturing;
6. the fit correction by functional thermic analysis;
7. the fit correction at the rugosity;
8. the changing from fit with unitary hole system (H/) to fit with unitary shaft system (/h), if is necessary;
9. the deviations showing on the diagram.

B.1.1. *The average clearance:* if the assemblage is in cylindrical shape:

$$v = \frac{\pi \cdot \phi \cdot n}{60} \quad \text{(m/s)}$$

where n is the rotation speed in rotations/minute (if Φ is in meters). In this way can be computed with [3] the average clearance like:

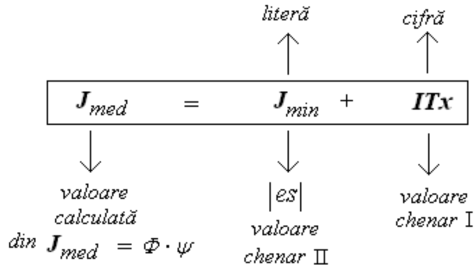
$$J_{med} = \phi \cdot \psi = \phi \cdot 0{,}8\sqrt[4]{\frac{\pi \cdot \phi \cdot n}{60}} \quad (\mu m) \qquad (4)$$

In the diagram (1) is presented table3 which, in border I, contains the standard tolerance IT01, IT0; IT1 … IT16 with grade of tolerance 01, 0, 1, 2 … 16 (recommended at special (finesses) mechanics to be from 1 to 10, and for the grade of tolerance in the machines building from 5 to 12 in generally). In the border II are presented the fundamental deviations of the shafts, for the 11 types of clearance fits: 8 fits named with a single letter (from a to h) and the other 3 fits named with two letters (cd, ef and fg).

The values $es \le 0$ from the border II will be, in absolute value, equal to the minimum clearance of the respective fit: $|es| = J_{min}$.

In the bottom side of the table 3 from diagram 1 is presented he places of the tolerance fields in function of the fit type and the meaning of the values which are written in borders I and II.

If : $T_D = T_d = ITx$ than can be computed the used "key of the abacus" from table 3 and [1]:



(5)

In [5] is established the "key of the abacus" composed for the clearance fits computing purpose on the ground of the unitary shaft.

In other words, the computed average clearance is equal to the minimum clearance (value from the border II of the table 3 in the absolute value : $|es|$, which on the column corresponds to a letter, and this letter will be the symbol of the designed fit type), plus a tolerance field ITx, which is founded in border I of the table 3 and this field is in correspondence to a number on the column. This symbol number represents the grade of tolerance (the grade of tolerance level) of the designed fit.

***Table 2** The clearance fits groups*

Class	The behavior of the twin pairs pieces				Examples
	Assembling	Running		Disassembling	
I-st	M	M		M	bearings, guides, mobile joints
II-nd	M	F*		M	toothed wheels mounted by wedges on the shafts, mobile casing on the tools-machine guides, fixed spanner
III-rd	M	(1)F*	(2)M*	M	(1) see class II examples; (2) balladeer toothed wheels
IV-th	M	F*		F*	assemblages soldered on the pieces circumference

Table 3

		TOLERANȚE FUNDAMENTALE ITx (μm)																		ABATERILE FUNDAMENTALE ALE ARBORILOR (μm)										
POZ.CÂMP DE TOL.		H01	H0	H1	H2	H3	H4	H5	H6	H7	H8	H9	H10	H11	H12	H13	H14	H15	H16	a	b	c	cd	d	e	ef	f	fg	g	h
PRECIZIA		01	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	Toate preciziile										
DIMENSIUNI NOMINALE (mm)	> 1... 3	0,3	0,5	0,8	1,2	2	3	4	6	10	14	25	40	60	105	140	250	400	600	-270	-140	-60	-34	-20	-14	-10	-6	-4	-2	0
	> 3... 6	0,4	0,6	1	1,5	2,5	4	5	8	12	18	30	48	75	120	180	300	480	750	-270	-140	-70	-46	-30	-20	-14	-10	-6	-4	0
	> 6... 10	0,4	0,6	1	1,5	2,5	4	6	9	15	22	36	58	90	150	220	360	580	900	-280	-150	-80	-56	-40	-25	-18	-13	-8	-5	0
	> 10... 14	0,5	0,8	1,2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100	-290	-150	-95	-69	-50	-32	-23	-16	-10	-6	0
	> 14... 18	0,5	0,8	1,2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100	-290	-150	-95	-85	-50	-32	-28	-16	-10	-6	0
	> 18... 24	0,6	1	1.5	2,5	4	6	9	13	21	33	62	84	130	210	330	520	840	1300	-300	-160	-110	-65	-65	-40	-28	-20	-12	-7	0
	> 24... 30																			-310	-170	-120	-100	-80	-50	-35	-25	-15	-9	0
	> 30... 40	0,6	1	1.5	2,5	4	7	11	16	25	39	62	100	160	250	390	620	1000	1400	-320	-180	-130								
	> 40... 50																			-340	-190	-140	-120	-100	-60	-42	-30	-17	-10	0
	> 50... 65	0,8	1,2	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900	-360	-200	-150								
	> 65... 80																			-380	-220	-170	-145	-120	-72	-51	-36	-21	-12	0
	> 80... 100	1	1,5	2,5	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200	-410	-240	-180								
	> 100... 120																			-460	-260	-200	-175	-145	-85	-60	-43	-25	-14	0
	> 120... 140																			-520	-280	-210								
	> 140... 160	1,2	2	3,5	5	8	12	18	25	40	63	100	160	250	400	630	1080	1600	2500	-580	-310	-230								
	> 160... 180																			-660	-340	-240	-210	-170	-100	-71	-50	-27	-15	0
	> 180... 200																			-740	380	260								
	> 200... 225	2	3	4,6	7	10	14	21	29	46	72	115	185	290	460	720	1150	1860	2900	-820	-420	-280								
	> 225... 250																			-920	-480	-300	-245	-190	-110	-78	-56	-31	-17	0
	> 250... 280	2,5	4	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200	-1050	-540	-330								
	> 280... 315																			-1200	-600	-360	-282	-210	-125	-88	-62	-33	-18	0
	> 315... 355	3	5	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600	-1350	-680	-400								
	> 355... 400																			-1500	-760	-440	-326	-230	-135	-96	-68	-37	-20	0
	> 400... 450	4	6	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000	-1650	-840	-480								
	> 450... 500																													

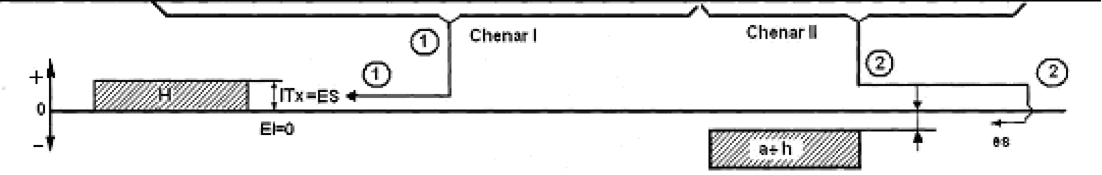


***Fig.1.** The places of the tolerance fields in function of the fits type (clearance case)*

The "Key" condition is: $|es| \approx ITx$ or $es > ITx$, and the best is that the two values to be closest.

Example : a sliding bearing with Φ = 8mm in diameter running with n = 10.000 rot/min, in class B1.

B.1.1 and **B.1.2.** *The designing of the fit type and the grade of tolerance levels:*

$$J_{med} = \phi \cdot \psi = \phi \cdot 0{,}8\sqrt[4]{\frac{\pi\phi n}{60}} =$$

$$= 8 \cdot 0{,}8 \cdot \sqrt[4]{\frac{\pi \cdot 0{,}008 \cdot 10000}{60}} = 9{,}15\mu m$$

We chose J_{med} = 9μm. It is established the "Key" formula [5].

$$\begin{array}{ccccc} & & g & & 4 \\ & & \uparrow & & \uparrow \\ 9 & = & 5 & + & 4 \\ \downarrow & & \downarrow & & \downarrow \\ J_{med} \text{ calculat} & & \text{valoare chenar II} & & \text{valoare chenar I} \end{array}$$

Taking into consideration the assemblage in the unitary hole fit system, the designed fit will be: $\phi 8\frac{H4}{g4}$. In [10] are presented the stages from B.1.3. to B.1.9.

B.2. In *the clearance fit designing process* are imposed the extreme (admissible) fit characteristics: J_{max} and J_{min}. In this case the designer task is more simple using table 3 again.

Taking into consideration the formula: $T_{aj} = J_{max} - J_{min} = ITx_D + ITx_d$; but accepting the simplification: $ITx_D = ITx_d = ITx$, results: $J_{max} = J_{min} + 2\ ITx$, and if it is known the values J_{max} and J_{min} than we can calculate ITx like : $ITx = \dfrac{J_{max} - J_{min}}{2}$.

In table 3 is selected from the border I the nearest value to the computed ITx , value which represents on the column the grade of tolerance group. Again, from table 3 - border II, it is selected the imposed $|es| = J_{min}$.

Example : for $\Phi 35$: $J_{max} = 52\mu m$; $J_{min} = 9\mu m$; $ITx = \dfrac{52-9}{2} = 21{,}5\mu m$ (value approximated to 25, and, in table 3 – border I, we will find the grade of tolerance 7) . $J_{min} = 9\mu m$ means that $|es| = J_{min}$ (from border II) for the shaft : „*g*".

It means that the fit is: $\phi 35\ H7/g6$.

The inference fit designing

The inference fits are chosen in the case of twin pairs parts which will achieve a inference (holding) state on the ground of the assemblage characteristic S (the inference which is achieved between the hole and the shaft).

In certain cases, where it is better to have inferences with small values, and the functional conditions involves a superior inference (holding) level to the level ensured by fit, for that achievement, at the inference fits too, there are equipped with afferent mechanical parts. In this way, the inference fits are classified in:

- inference fits with holding by control ;
- inference fits with holding by control with inference level compensation.

The inference fits can be assembled in these variants:

- "cool" pressed inference fits ;
- "warm" assembled inference fits (milled - in the heating hole case).

For the cool pressed inference fits, the rugosity on the pieces surface will be smoothed, the effective inferences after assemblage will be decreased, and because these don't give an exactly information about the inference level, and because these depends on the assemblage diameter too (ϕ=d), in [1] is defined the relative effective average inference like:

$$S_{medER} = \frac{S_{medE}}{\phi} = \frac{S_{med} - c}{\phi}; \qquad S_{med} = \frac{S_{max} + S_{min}}{2} \tag{6}$$

where, more than the well known symbols, the symbol "c" is a variable which depends on several factors: the rugosity value and shape, the inference value, the holes width, the hole overlay length over the shaft, the elasticity absolute values of the twin pieces stuff etc.

In function of the relative effective inference (minimum, average etc.), in [1] the inference fits are classified in four groups:

1. *very hard* inference fits, with S_{minER} = 1µm/mm (1‰);
2. *hard* inference fits, with S_{medER} = 1µm/mm (1‰);
3. *medium* inference fits, with S_{medER} = 0,5µm/mm (0,5‰);
4. *light* inference fits, with S_{medER} = 0,25µm/mm (0,25‰).

At the inference fits designing, he designer can meet two situations:

A. The assemblage technical and functional information are given;

B. The inference extreme characteristics are imposed (S_{max} and S_{min}).

A. The inference fits from the first case are designed by computing the minimum inference, following these stages:

A1. the fit type designing;
A2. the grade of tolerance levels designing;
A3. the deviations computing;
A4. the assemblage characteristic computing;
A5. the computing of the manufacturing fit control;
A6. the computing of the rugosity fit control (except for the warm assembled inference fit);
A7. the fit control check-up by thermic analysis;
A8. the designed fit change from a grade of tolerance type to other (in case from H/ to /h or vice versa);
A9. the fit check-up at the maximum inference and σ_c (the flow limit of the stuff);
A10. the deviations showing on the diagram.

In table 4 there are presented the standard tolerances (border I) for the grade of tolerance levels 01, 0, 1, 2 … 16 and the fundamental inference deviations of the shafts (border II).

From fig.2 (attacked to the table 4) is established the "Key" used for abacus [1]. In [6] is established the "Key" used for abacus designed for the control system with unitary shaft base.

The linking formula (if $ITx_D = ITx_d = ITx$) is:

$$\underset{\substack{\text{valoare}\\ \text{calculată}}}{S_{min}} = \overset{\text{literă}}{\underset{\substack{\text{tabelul 4}\\ \text{chenar II}}}{ei}} - \overset{\text{cifră}}{\underset{\substack{\text{tabelul 4}\\ \text{chenar I}}}{ITx}} \tag{7}$$

In other words, the calculated minimun interference is equal to the lower deviation (see value in the 2nd frame of table 4) which corresponds vertically to a letter, this will the be symbol of the projected fit, minus a tolerance zone *ITx,* which can be found in the 1st frame of table for and to whom a digit corresponds vertically. This will be the symbol digit which represents the precision class (precision) of the projected fit. Example: Given the following data desing the fit type from the interference group, for assembling a cogwheel on a tubular shaft (fig 3):

$d = \Phi = 10$mm; $d_1 = 5$mm; $d_2 = 20$mm;
$\ell = 10$mm; $P = 180$daN; $\mu = 0{,}08$; $\upsilon = 0{,}3$;
$E_A = 2{,}1 \cdot 10^4$daN/mm^2 $= E_B$;

Table 4

Chenar I — TOLERANȚE FUNDAMENTALE ITx (μm)

POZ.CÂMP DE TOL.	H01	H0	H1	H2	H3	H4	H5	H6	H7	H8	H9	H10	H11	H12	H13	H14	H15	H16
PRECIZIA	01	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
DIMENSIUNI NOMINALE (mm) > 1... 3	0,3	0,5	0,8	1,2	2	3	4	6	10	14	25	40	60	106	140	250	400	600
> 3... 6	0,4	0,6	1	1,5	2,5	4	5	8	12	18	30	48	75	120	180	300	480	750
> 6... 10	0,4	0,6	1	1,5	2,5	4	6	9	15	22	36	58	90	150	220	360	580	900
> 10... 14	0,5	0,8	1,2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100
> 14... 18	0,5	0,8	1,2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100
> 18... 24	0,6	1	1,5	2,5	4	6	9	13	21	33	52	84	130	210	330	520	840	1300
> 24... 30																		
> 30... 40	0,6	1	1,5	2,5	4	7	11	16	25	39	62	100	160	250	390	620	1000	1600
> 40... 50																		
> 50... 65	0,8	1,2	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900
> 65... 80																		
> 80... 100	1	1,5	2,5	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200
> 100... 120																		
> 120... 140	1,2	2	3,5	5	8	12	18	25	40	63	100	160	250	400	630	1080	1600	2500
> 140... 160																		
> 160... 180																		
> 180... 200	2	3	4,5	7	10	14	21	29	46	72	115	185	290	460	720	1150	1860	2900
> 200... 225																		
> 225... 250																		
> 250... 280	2,5	4	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200
> 280... 315																		
> 315... 355	3	5	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600
> 355... 400																		
> 400... 450	4	6	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000
> 450... 500																		

Chenar II — ABATERILE FUNDAMENTALE ALE ARBORILOR (μm)

POZ.CÂMP DE TOL.	m	n	p	r	s	t	u	v	x	y	z	za	zb	zc
PRECIZIA	Toate preciziile													
DIMENSIUNI NOMINALE (mm) > 1... 3	+2	+4	+6	+10	+14	-	+18	-	+20	-	+26	+32	+40	+60
> 3... 6	+4	+8	+12	+15	+19	-	+23	-	+28	-	+35	+42	+50	+80
> 6... 10	+6	+10	+15	+19	+23	-	+28	-	+34	-	+42	+52	+67	+97
> 10... 14	+7	+12	+18	+23	+28	-	+33	-	+40	-	+50	+64	+90	+130
> 14... 18	+7	+12	+18	+23	+28	-	+33	+39	+45	-	+60	+77	+108	+150
> 18... 24	+8	+15	+22	+28	+35	-	+41	+47	+54	+63	+73	+98	+136	+188
> 24... 30						+41	+48	+55	+64	+75	+88	+118	+160	+218
> 30... 40	+9	+17	+26	+34	+43	+48	+60	+68	+80	+94	+112	+148	+200	+274
> 40... 50						+54	+70	+81	+95	+114	+136	+180	+242	+325
> 50... 65	+11	+20	+32	+41	+53	+66	+87	+102	+112	+144	+172	+226	+300	+405
> 65... 80				+43	+59	+75	+102	+120	+146	+174	+210	+274	+360	+480
> 80... 100	+13	+23	+37	+51	+71	+91	+124	+140	+175	+214	+258	+335	+445	+585
> 100... 120				+54	+79	+104	+144	+172	+210	+254	+310	+400	+525	+690
> 120... 140	+15	+27	+43	+63	+92	+122	+170	+202	+248	+300	+365	+400	+620	+800
> 140... 160				+65	+100	+134	+190	+228	+280	+340	+345	+470	+700	+900
> 160... 180				+68	+108	+146	+210	+252	+310	+380	+415	+535	+780	+1000
> 180... 200	+17	+31	+50	+77	+122	+166	+236	+284	+350	+425	+520	+670	+880	+1150
> 200... 225				+80	+130	+180	+258	+310	+365	+470	+575	+740	+960	+1250
> 225... 250				+84	+140	+196	+284	+340	+425	+520	+640	+820	+1050	+1350
> 250... 280	+20	+34	+56	+94	+158	+218	+315	+385	+475	+580	+710	+920	+1200	1550
> 280... 315				+98	+170	+240	+350	+425	+525	+650	+790	+1000	1300	+1700
> 315... 355	+21	+37	+62	108	+190	+268	+390	+496	+590	+730	+900	+1150	1500	+1900
> 355... 400				+114	+208	+294	+435	+530	+660	+820	+1000	+1300	+1650	+2100
> 400... 450	+23	+40	+68	+126	+232	+330	+490	+595	+740	+920	+1100	+1450	+1850	+2400
> 450... 500				+132	+252	+360	+540	+640	+820	+1000	+1250	+1600	+2100	+2600

Fig.2 *Tolerance zones depending on the type of the fit (for intereference fit)*

$$p = \frac{P}{\pi d \ell \mu} = \frac{180}{\mu \cdot 10 \cdot 10 \cdot 0{,}08} = 7{,}16 \text{ daN/mm}^2,$$

or in the case were the torque is known (T), for example T = 900daNmm,from $T = P\frac{d}{2} = P\frac{10}{2} = P \cdot 5$ daN/mm), which replaced in the formulae of pressure:

$$p = \frac{2T}{\pi d^2 \ell \mu} = \frac{2 \cdot 900}{\pi \cdot 10^2 \cdot 10 \cdot 0{,}08} = 7{,}16 \text{ daN/m}^2$$

$$k_A = \frac{1+\left(\frac{d_1}{d}\right)^2}{1-\left(\frac{d_1}{d}\right)^2} - \upsilon = \frac{1+\left(\frac{5}{10}\right)^2}{1-\left(\frac{5}{10}\right)^2} - 0{,}3 =$$

$$= \frac{1+0{,}25}{1-0{,}25} - 0{,}3 = 1{,}3666$$

$$k_B = \frac{1+\left(\frac{d}{d_2}\right)^2}{1-\left(\frac{d}{d_2}\right)^2} + \upsilon = \frac{1+\left(\frac{10}{20}\right)^2}{1+\left(\frac{10}{20}\right)^2} + 0{,}3 =$$

$$= \frac{1+0{,}25}{1-0{,}25} + 0{,}3 = 1{,}9666$$

$$S_{min} = 7{,}16 \cdot 10 \cdot \left(\frac{1{,}3666}{2{,}1 \cdot 10^4} + \frac{1{,}9666}{2{,}1 \cdot 10^4}\right) =$$

$$= 7{,}16 \cdot 10 (0{,}00006508 + 0{,}00009365) =$$

$$= 0{,}01136\, mm = 11{,}36\, \mu m$$

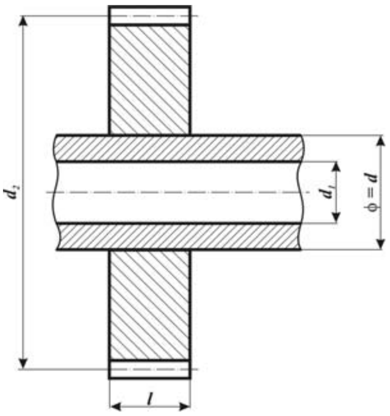
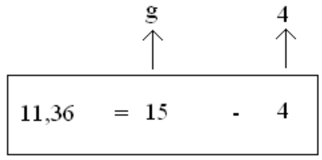


Fig.3 *Assembly of a cogwheel on a tubular shaft*

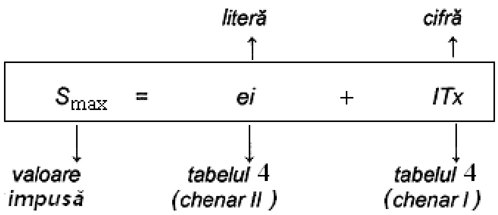
A.1; A.2; Determining the fit type and precision classes. Applying formula (7) for the given conditions:



So the fit is $\Phi 10H4\ /\ p4$. The steps A4÷A10 are detailed in [10].

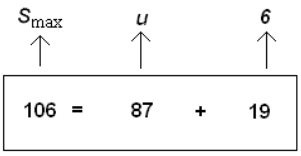
B. For the interference fit the following limits are set S_{max} şi S_{min}. The problem is solved using Table 4.

From S_{max} set as: $S_{max} = es - EI = es - o$ (in basic hole, see fig.2), but $es = ei + ITx$, thus $S_{max} = ei + ITx$.and $S_{min} = ei - ES = ei - ITx$



(8)

Example: $S_{max} = 106\mu m$; $S_{min} = 68\mu m$ for $\Phi = 55mm$. Using formula (8) in the given example:



The resulting fit is $\phi 55\frac{H6}{u6}$; with the following deviations: $EI = 0$; $ES = 19\ \mu m$; $ei = 87$; $es = 106\mu m$.

Transition fits design

Transition fits (transitional or uncertain) make up a proportion which tends towards 1:

$$\frac{D}{d} \to 1 \text{ because } D \leftrightarrow d \qquad (9)$$

through this being created the conditions needed for clearance to appear and also interference in a certain percentage

As both clearance and interfence are relatively small (see formula 9), the excentricity (deviation from the position) of the geometrical axes of the two parts is small, towards 0, therefor we obtain concentrical assemblies, thus the conclusion that the transition fits are chosen to ensure a high assembly precision. Transition fits having, both, clearance and interference the insurance against relative movement is done through braces, with the exception of the tolerable fit (*H/j*) which in certain cases can be mobile.

In the diagram presented in fig 4 are depicted the variations of the assembly precision, the degree of stiffness and mobility, based on the assembly values (clearance or interference), for the groups and types of fit, in the interval (50...80]mm for the nominal diameter, for the tolerance steps indicated in [8].

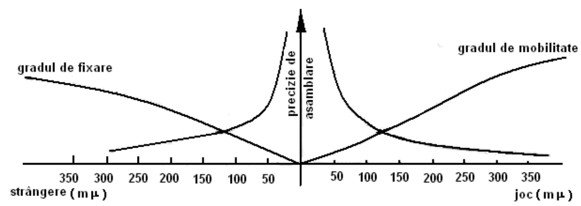


Fig.4 *The variation of the assembly precision, the degree of mobility and stiffness based on the assembly characteristics*

Given the formulas from [1]:

$$\begin{aligned} &J = 2e; J_{max} = 2e_{max};\\ &J_{min} = 2e_{min};\\ &S = -2e; S_{max} = -2e_{min};\\ &S_{min} = -2e_{max} \end{aligned} \qquad (10)$$

for transition fits(with clearance) results that:

$$\begin{aligned} T_{J/S} &= J_{max} - J_{min} =\\ &= J_{max} + S_{max} =\\ &= 2e_{max} + (-2e_{min}) = 2T_e \end{aligned} \qquad (11)$$

And for transition fits (with interference):

$$\begin{aligned} T_{J/S} &= S_{max} - S_{min} =\\ &= S_{max} + J_{max} =\\ &= -2e_{min} + 2e_{max} = 2T_e \end{aligned} \qquad (12)$$

where T_e – excentricity tolerance.

The kinds of transition fits (including the formulaes to calculate (deducted by the author of [1]) e_{max}) are presented in table no. 5 (Table 3.17 from [1]).

The fits from types **1** and **2** are used for assemblies that are subjected to frequent assemble/dis-assemble operations and those from type **3** for assemblies that are subjected to less frequent assemble/dis-assemble operations. Fits from type **4** are used in assemblies that are used for high static/dynamic loads, and are only disassembled in capital maintenances, and those from the **5**th kind are used for assemblies using a cleat, when they are going to be overly braced.

When designing transition fits, knowing the nominal diameter or the assembly, the following data arrises [1]:

1 - The instructions refering to the assembly precision and the assembly/dis-assembly frequency are known.

2 - The majoritary maximum characteristic is set:

a - J_{max} când $J_{max} > S_{max}$;

b - S_{max} când $S_{max} > J_{max}$.

3 - The majoritary maximum characteristic is set, as above, and the standard tolerance (T_{aj}) or the average square deviation (σ).

4 - % is set vor the majoritary maximum characteristic ($\%J_{max}$ sau $\%S_{max}$) and t(sau he variation of the fit (or σ).

Table 5

Transition family	Assembly Characteristics J; S ei	Excentricity „e” [μm] Parameter for assembly imprecission			Mounting precision
		Processing precision			
		a	B	c	
		High	Medium	Satisfactory	
1	2	3a	3b	3c	4
1.	$J_{max} » S_{max}$ Jocuri; $J > 90\%$ $ei < 0$	Φ (1…500)mm $e_{max} = 2{,}5\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 3{,}6\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 5{,}9\sqrt[3]{\phi_m}$	Modest
2.	$J_{max} > S_{max}$ Jocuri; $J > 80\%$ $ei < 0$	Φ (1…120)mm $e_{max} = 2{,}5\sqrt[3]{\phi_m}$	Φ (1…80)mm $e_{max} = 3\sqrt[3]{\phi_m}$	Φ (1…80)mm $e_{max} = 5\sqrt[3]{\phi_m}$	Satisfactory
		Φ (120…315)mm $e_{max} = 2{,}5\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{2}$	Φ (80…315)mm $e_{max} = 3\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{2{,}5}$	Φ (80…180)mm $e_{max} = 5\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{2{,}5}$	
		Φ (315…500)mm $e_{max} = 2{,}5\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{2{,}5}$	Φ (315…500)mm $e_{max} = 3\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{1{,}5}$	Φ (180…500)mm $e_{max} = 5\sqrt[3]{\phi_m} + \frac{\sqrt{\phi_m}}{1{,}5}$	
3.	$J_{max} = S_{max}$ Joc ≈ 50% Strâng. ≈50% ei=0 sau (ei=0+ε)	Φ (1…500)mm $e_{max} = 1{,}4\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 2\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 3{,}75\sqrt[3]{\phi_m}$	High
4.	$S_{max} > J_{max}$ Strâng. >80% $e\,i > 0$	Φ (1…500)mm $e_{max} = 0{,}22\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 0{,}97\sqrt[3]{\phi_m}$	Φ (1…500)mm $e_{max} = 2{,}4\sqrt[3]{\phi_m}$	Huge
5.	$S_{max} » J_{max}$ Strâng →100% $e\,i > 0$	$e_{max} < 0$; $J < 0$ aj. cu strângere	$e_{max} < 0$; $J < 0$ aj. cu strângere	Φ (1…500)mm $e_{max} = 1{,}3\sqrt[3]{\phi_m}$	[Pref.I şi II]: very huge
Choice criteria:→		Second preference	First preference	Third preference	

In [1] the algorythm steps for designing the type of the fit using the “key” proposed by the author.

If in [1] the design algorythm of the type of fit based on basic hole is presented, in [3] the design of the transition fit based on basic shaft is presented.

Calculus Example

1. Design the fit for a cogwheel used assembled on the shaft of a gearbox (Φ = 75mm).

For the given example is demanded the realisation of the montage with small forces.

From Table no. 5 the **2**nd kind is considered ($J_{max} > S_{max}$; J > 80%), choice **I** (for Φ = 1...80), $\left(e_{max} = 3\sqrt[3]{\phi_m}\right)$,

$$\phi_m = \sqrt{50} \cdot \sqrt{80} = 63{,}20 \cong 63 \text{ mm},$$

$$e_{max} = 3\sqrt[3]{\phi_m} = 3\sqrt[3]{63} = 3 \cdot 3{,}98 = 11{,}94 \,\mu\text{m}$$

knowing the value of the maximum excentricity, you can determine the maximum clearance: $J_{max} = 2e_{max} = 2 \cdot 11{,}94 = 23{,}88\mu m = 24\mu m$

Using one of the fundamental formulas $J_{max} = ES - ei$, and using fig. 4 and 6th table, you can determine: $J_{max} = ES - ei = ITx - ei$.

Considering $ITx_d = ITx_D = ITx$ şi $ei<0$ in the “key” formula for using the nomogram is:

$$\underbrace{J_{max}}_{\substack{valoare\\ calculata}} = \underbrace{ITx}_{\substack{valoare\\ chenarI}} - \underbrace{(-ei)}_{\substack{valoare\\ chenarII}} \qquad (13)$$

In other words, the maximum clearance is equal to the tolerance zone *Itx,* which is found in frame I of the 6th Table, and for which a digit corresponds vertically that will represent the grade of tolerance(class of precision, tolerance step) of the projected fit minus the lower deviation (with $ei<0$) (value from the frame II from table 6, which corresponds vertically to a letter), this will be the symbol letter for the type of projected fit.

For J_{max} = 24μm, you can create the “key” formula:

$$24 \approx \underbrace{\overset{\overset{6}{\uparrow}}{19}}_{\substack{\downarrow\\ \text{valoare}\\ \text{chenar I}}} - \underbrace{\overset{\overset{j}{\uparrow}}{(-7)}}_{\substack{\downarrow\\ \text{valoare}\\ \text{chenar II}}}$$

Considering the assembly in the fit system with basil hole, the projected fit will be $\Phi 75H6\,/\,j5$. In machinery construction transition fit and precision classes 5,6 and 7 (with the following values IT5, IT6 and IT7)are exclusively used .

It’s worth mentioning that the “key” formula above could be fulfilled by other combinations of values, but you will choose the variant left after eliminating all the incompatibilities (between the precision class of the hole and that of the shaft there has to be a parity or a difference of maximum ±1), and he existence of the parity of the smallest difference between the value previously determined and that resulted by using the “key” formula (13). In [10] there are calculus examples for 2); 3); 4).

Table 6

	TOLERANȚE FUNDAMENTALE ITx (μm)																		ABATERILE FUNDAMENTALE ALE ARBORILOR (μm)										
POZ.CÂMP DE TOL.	IT01	IT0	IT1	IT2	IT3	IT4	IT5	IT6	IT7	IT8	IT9	IT10	IT11	IT12	IT13	IT14	IT15	IT16	js	j			k			m	n	p	r
PRECIZIA	01	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16		5/6	7	8	3	4-7	8				
DIMENSIUNI NOMINALE (mm) <3	0,3	0,5	0,8	1,2	2	3	4	6	10	14	25	40	60	100	140	250	400	600	ABATERILE LIMITĂ SUNT EGALE CU ± 1/2ITx	-2	-4	-6	0	0	0	+2	+4	+6	+10
> 3... 6	0,4	0,6	1	1,5	2,5	4	5	8	12	18	30	48	75	120	180	300	480	750		-2	-4		0	+1	0	+4	+6	+12	+15
> 6... 10	0,4	0,6	1	1,5	2,5	4	6	9	15	22	36	58	90	150	220	360	580	900		-2	-5		0	+1	0	+6	+10	+15	+19
> 10... 14 > 14... 18	0,5	0,8	1,2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100		-3	-6		0	+1	0	+7	+12	+18	+23
> 18... 24 > 24... 30	0,6	1	1.5	2,5	4	6	9	13	21	33	52	84	130	210	330	520	840	1300		-4	-8		0	+2	0	+8	+15	+22	+28
> 30... 40 > 40... 50	0,6	1	1.5	2,5	4	7	11	16	25	39	62	100	160	250	390	620	1000	1600		-5	-10		0	+2	0	+9	+17	+26	+34
> 50... 65 > 65... 80	0,8	1,2	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900		-7	-12		0	+2	0	+11	+20	+32	+41 / +43
> 80... 100 > 100... 120	1	1,5	2,5	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200		-9	-15		0	+3	0	+13	+29	+37	+51 / +54
> 120... 140 > 140... 160 > 160... 180	1,2	2	3,5	5	8	12	18	25	40	63	100	160	250	400	630	1000	1600	2500		-11	-18		0	+3	0	+15	+27	+43	+63 / +65 / +68
> 180... 200 > 200... 225 > 225... 250	2	3	4,5	7	10	14	21	29	46	72	115	185	290	460	720	1150	1850	2900		-13	-21		0	+4	0	+17	+31	+50	+77 / +80 / +84
> 250... 280 > 280... 315	2,5	4	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200		-16	-26		0	+4	0	+20	+34	+56	+94 / +98
> 315... 355 > 355... 400	3	5	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600		-18	-28		0	+4	0	+21	+37	+62	+108 / +114
> 400... 450 > 450... 500	4	6	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000		-20	-32		0	+5	0	+23	+40	+68	+126 / +132

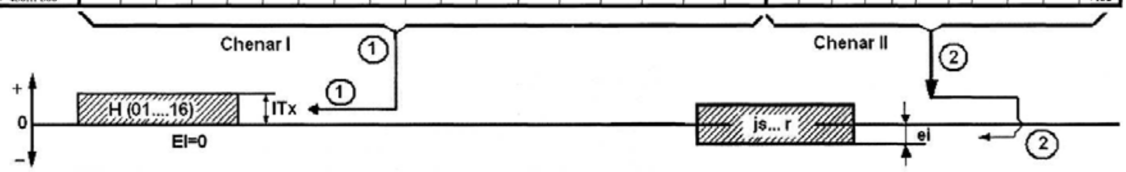


Fig.5 *Location field of tolerance depending on the fits (transition fits)*

Conclusions

If the usage of a standardized fit for which calibers exist [2] is recommended for small volume production, for large volumes production, if the parts are verified based on calibers then it’s worth creating non-standard calibers.

Transforming from the fit system using basic shaft to the fit sistem using basic hole in the case that the two precision steps are not equal, the part woult retain its grade of tolerance. With the purpose of eliminating routine tasks(simple math operations, consulting deviation tables, fundamental tolerances etc.) and substantially reducing the time required for these operations, the problem of fit designing can be easily solved using an electronic calculator[1],[9].

After chosing from the fit list the most convenient one, the limits of deviation and the assemply characteristics will be determined.

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