



## Optimal dimensioning of the transmission shaft of a longitudinal gearbox

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### Abstract

#### Purpose:

The purpose of the paper is to accomplish the analysis of a transmission shaft belonging to a longitudinal gearbox in order to reduce the axial clearances of the parts assembled on the shaft. The paper demonstrates also that the study of assembly conditions allows simplifying the analysis and the evaluation of different dimensioning schemes.

#### Method:

A previous functional approach to optimal dimensioning is used but it is associated to the analysis of relationships between the parts assembled on the shaft, the corresponding functional surfaces and the related dimensioning schemes. Then, a statistical analysis of tolerance chain is accomplished by using different IT grades that characterize dimensional tolerances.

#### Result:

The optimal functional dimensioning scheme of an automotive transmission shaft is accomplished. The scheme is able to guarantee the minimum axial clearances of parts assembled on the shaft.

#### Discussion & Conclusion:

The paper deals with the application of a functional approach to the optimal dimensioning of an automotive transmission shaft. The automotive transmission set is characterized by different working conditions, geometrical constraints and design parameters. Therefore, a preliminary decomposition of the assembly in several subsystems or groups is necessary to clearly analyse the longitudinal gearbox. Then, the study of assembling conditions and related functional surfaces allow simplifying the evaluation of different dimensioning schemes. In fact, the functional approach, aimed to optimal dimensioning, uses a worst-case and statistical analyses to directly check the axial clearances of the wheels assembled on the shaft. Therefore, the effects of dimensioning are directly evaluated in terms of performances of the transmission. Therefore, the different dimensioning schemes are evaluated through the functional approach, as the optimal scheme is able to guarantee the whole set of functional requirements of the shaft.

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## 1 Introduction

Researchers engaged in the product-process design are very attentive to the issue of dimensioning and tolerancing. For over 75 years, the method of Geometric Dimensioning and Tolerancing (GD&T) was studied and the activities related to dimensioning and tolerancing experimented a long evolution [7], as in the case of Computer Aided Tolerancing (CAT) software tools [13]. Many authors focused on the definition of algorithms, methods and tools to evaluate optimized tolerances and obtain minimum clearances [7]. Ngoi and Min [5] proposed an algorithm for optimum tolerance allocation. Still recently, authors in [11],[12] defined some rules to improve the dimensioning process. Other authors, as Chavanne et al. [1], defined the CLIC method and Jamil et al. [6] defined CAD integrated tools to achieve automatic dimensioning of parts, by using automatic feature recognition. Several years ago, Serrano [14] proposed a theoretical approach, based on graphs, for

the automatic generation of geometrical constraints. Maziero [11] described a method for automatic dimensioning of parts characterized by cylindrical features. The method was essentially based on functional dimensioning. Also Chen [3] proposed an approach based on feature extraction. Few authors have focused on how to evaluate different dimensioning schemes with synthetic indexes [12] in order to individuate the optimal scheme according to the requirements to be met in a simplified way. Lanzotti et al. [7] proposed a functional approach to optimal dimensioning; in particular, the dimensions and tolerances are determined by satisfying all relationships represented in the Functional Requirements/Dimensions (FR/D) Matrix. The authors do not perform the nominal value optimization as in Parkinson [4], but intend to extend Parkinson's method to the definition of the dimensioning scheme that fulfils the FRs of the assembly.

When such method allows satisfying the assembly requirements, the Suh's theory is certainly useful [15] and it is possible to refer to *Functional Dimensioning*.

One of the critical issues is to locate the Functional Requirements (FRs) and convert them into dimensional specifications. In particular, some authors developed also software to convert FRs of a product into dimensional specifications by using a FR/D Matrix [1].

This paper applies the approach proposed in [7] to a shaft of an automotive longitudinal gearbox with the focus on the shift lever movement [9][10]. The main aim is to evaluate the approach applicability to a specific gearbox.

Section 2 presents a brief overview of the approach. Section 3 describes the gearbox and summarizes the characteristics related to the dimensioning problem; section 4 illustrates the application of the approach to the Functional Dimensioning to the *Group 2* of the gearbox. In section 5 and section 6 the optimal dimensioning scheme and statistical analysis are evaluated, respectively. Finally, Section 7 draws the conclusions.

## 2 Approach overview

The functional approach for optimal dimensioning, used in this paper is proposed in [7]. It provides a new formulation related to mechanical assemblies characterized by a series of different configurations, repetitively assumed by the components of the assembly.

The methodology allows to obtain the dimensioning that guarantees the fulfillment of a set of Functional FRs at the minimum manufacturing costs. In particular, the general expression for FRs is:

$$[FR] = [CM][D] \quad (1)$$

where  $[CM]$  is the chain matrix,  $[FR]$  is the Functional Requirement array and  $[D]$  is the Dimension array.

If we now consider the case of mechanical assemblies operating between two or more working conditions, repetitively assumed, we could introduce an expression for the total *Information Content*  $I_{tot}$  of the dimensioning, that has to be minimized.

$$I_{tot} = I_D + I_{FR} \quad (2)$$

$I_D$  represents the *Information Content* due to nominal dimensions that influence the FRs and the corresponding tolerance intervals, as it follows.

$$I_D = C \cdot \sum_{i \in \Phi} \frac{V_i}{\Delta_i} \quad (3)$$

$$\text{subject to: } \sum_{i \in \Phi_{FRj}} \Delta_i \leq \Delta_{FRj} \quad j = 1 \dots n_{FR}$$

where:

- $V_i$  is the  $D_i$  nominal value;
- $\Phi$  is the set of dimensions that influence the whole set of FRs;
- $C$  is a constant value (for numerical applications in this paper  $C=0.1$ , when  $V_i$  and  $\Delta_i$  are expressed in *mm*);
- $\Phi_{FRj}$  is the set of dimensions that influences each  $FR_j$ ;
- $\Delta_i$  is the tolerance interval i.e. the accepted variability for  $D_i$  in relation to  $V_i$ ;
- $\Delta_{FRj}$  is the accepted variability for  $FR_j$ .

$I_{FR}$  represents a loss of *Information Content* due to the incomplete decoupling between Functional Requirements and dimensions.

First, the initial design problem is decomposed in a set of ideal design problems in which the Information Content  $I_{FR}$ , related both to each Functional Requirement and a dimensioning scheme, is the minimum one. Then, the variation of the dimensional value associated to each Functional Requirement is taken into account, when other dimensioning schemes are used.

$$I_{FR} = \sum_J K_{mJ} * \frac{\sqrt{(\Delta_{FR_{mJ}} - \Delta_{FR_{J \min}})^2}}{\Delta_{Jm}} \quad \forall m \quad (4)$$

where:

- $m$  is the number of assembly working conditions;
- $J$  is the total number of FRs related to the set of working conditions;
- $\Delta_{Jm}$  is the tolerance range of the dimensions related to the  $J$ -th FR and the  $m$ -th dimensioning scheme;
- $K_{mJ}$  is the number of dimensions belonging to the tolerance chain related to  $J$ -th FR and  $m$ -th dimensioning scheme;
- $\Delta_{FR_{mJ}}$  is the tolerance of the dimensional value associated to the  $J$ -th FR within the  $m$ -th dimensioning scheme;
- $\Delta_{FR_{J \min}}$  is the minimum value of the  $\Delta_{FR_{mJ}}$ .

The functional dimensioning scheme is obtained by minimizing the  $I_{tot}$  index (eq. 5), subject to the complete set of constraints.

$$\begin{aligned} \min! \quad & I_{tot} \\ \text{subject to: } \quad & \sum_{i \in \Phi_{FRj}} \Delta_i \leq \Delta_{FRj} \quad j = 1 \dots n_{FR} \end{aligned} \quad (5)$$

## 3 Application to a transmission shaft

The proposed approach is applied to a shaft of an automotive transmission set (fig. 1). In the following, the transmission set is described, the different working conditions are analyzed and the functional dimensioning scheme for the transmission shaft is proposed. Finally, the axial clearances, that influence the performances of the shaft, are evaluated by using a worst-case and statistical approach.

### 3.1 Gearbox architecture and functionality

The gearbox, made by an automotive Asiatic company, has got a longitudinal layout. The transmission set is made by three shafts. In particular, the set is composed by a main shaft, an auxiliary shaft and, finally, by a transmission shaft (fig. 1). The axes of shafts are coplanar. The main shaft and the transmission shaft are coaxial and are coupled with a radial bearing and an axial bearing; the shafts have an angular relative motion. Both shafts are fixed to the gearbox casing with one bearing. The auxiliary shaft engages with both shafts to realize the opportune speed ratios except for the fourth gear which is realized when angular relative motion of main and transmission shaft is constrained. The auxiliary shaft do not take part to the motion transmission in this case.

The auxiliary shaft has all fixed wheels while the transmission shaft has all neutral wheels. The motion comes from the main shaft by its single fixed wheel, then through the auxiliary shaft is transmitted to the transmission shaft and then to the differential.

Since the auxiliary shaft is fixed to the case by bearings and the wheels are fixed on it, they never move, while all wheels on the transmission shaft can be moved in axial direction.

All wheels are with helical toothed and are assembled on the shafts with an axial clearance, in order to ensure the proper functioning. During the working, due to helical teeth, each wheel is subjected to the action of two orthogonal forces: a tangent force that gives the rotational motion; an axial force that causes an axial displacement. The direction of this axial movement depends by working conditions, acceleration or deceleration phase.

During the acceleration phase, the engine transmits the motion to the wheels. In such condition, the driving gear is on the auxiliary shaft while the driven gear is on the transmission shaft. The direction of rotation of the wheels, the relative positions of wheels and the inclination of the helical teeth, define the verses of the resulting axial forces [15]. Since on the transmission shaft all wheels are driven, engaged and not engaged gear wheels move in the same direction.

During the deceleration phase, the wheels drag the engine. In particular, the side of the teeth under pressure changes. Also the axial force direction, acting on the engaged wheels during the acceleration phase, changes. The not engaged wheels located on the transmission shaft are still driven so they don't change the position because they are pushed towards opposite axial verse with respect to the engaged gear.

The axial movements are only made by the wheels on the transmission shaft for which reason the present study deals the transmission shaft.

The first, second, third, fifth and reverse gear wheels, named 1, 2, 3, 5, R, respectively, can run neutrally respect to the shaft, due to the interposition of roller cages. These wheels are integral with the shaft rotation only when the corresponding gear is engaged. Then, between the first and second wheels, between the fifth and reverse wheels and to the left of the third wheel is assembled the first-second wheel-hub, fifth-reverse wheel-hub, third-fourth wheel-hub, respectively. Those elements are integral with the shaft rotation due to a grooved profile. Each wheel-hub in axial direction is fixed by a ring that forces it in its finally position. The particular geometry of the ring slot (fig. 2) deforms the ring that applies an axial force on wheel-hub that is pushed on the shaft shoulder, so the wheel-hub in axial direction is fixed. This configuration (fig. 4) for any wheel-hub is repeated as depicted in fig. 3.

The assembly is completed by means of a bearing B. The dashed line L in fig. 3 represents the lateral side of the ring that locates the bearing respect to the gearbox casing. Therefore, the dashed line L is fixed i.e. it does not depend by tolerances assigned to the shaft or by working conditions of the shaft. Since the wheel-hubs are fixed, the wheel axial movements are very limited. In fact every wheel is limited in its axial movement from a side by wheel-hub and the other side by a shoulder.

The subsystem was divided in three functional groups (fig. 3):

- Group 1: 3 and 4 wheels and wheel-hub H<sub>3-4</sub>;
  - Group 2: 1 and 2 wheels and wheel-hub H<sub>1-2</sub>;
  - Group 3: 5 and R wheels and wheel-hub H<sub>5-R</sub>;
- which correspond to three mechanics guided by the lever. Due to the similarities between the three groups only the analysis related to the group 2 will be presented.

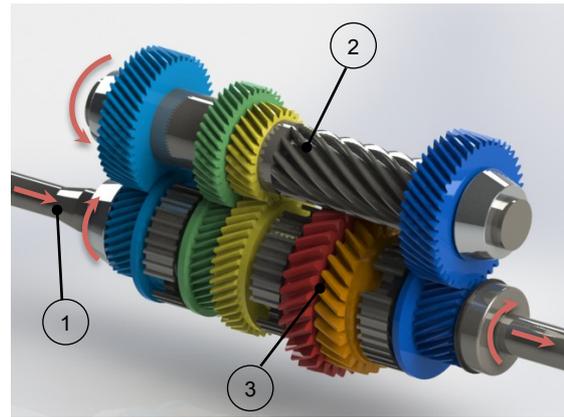


Fig. 1 CAD model simplified of the transmission set. The red arrows represent the transmission of the motion during the acceleration phase: 1) main shaft; 2) auxiliary shaft; 3) transmission shaft.

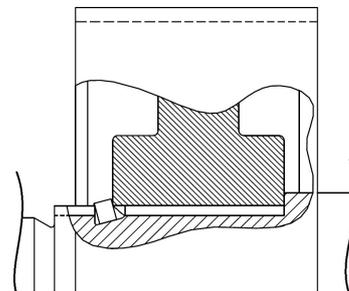


Fig. 2 Section view of the wheel-hub configuration depicting the H<sub>1-2</sub> case.

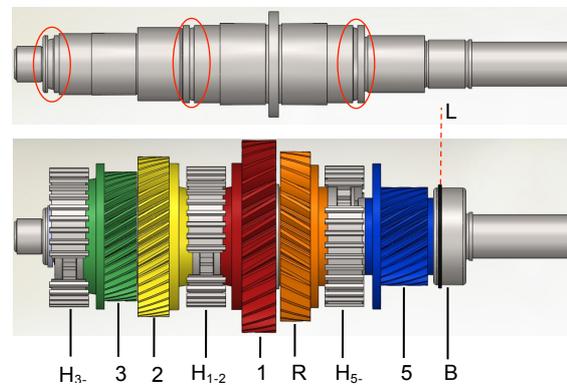


Fig. 3 (a) CAD model of the transmission shaft; (b) CAD model of the subassembly "transmission shaft".

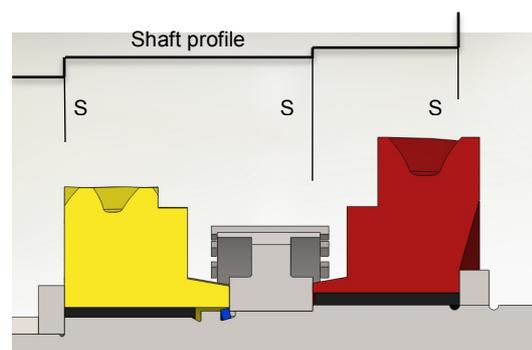


Fig. 4 Section view of the Group 2.

### 3.2 Functional group2

The functional Group 2 is composed by:

- 1) first gear wheel ,
- 2) second gear wheel,
- 3) first-second wheel-hub.

and transmission shaft more, but this is true for all groups.

From the analysis of the subassembly (fig. 4), it is possible to define three functional surfaces: S1, due to a ring fixed in its position by a spacer; S2 and S3, due to the shaft shoulders.

## 4 Functional dimensioning schemes related to Group 2

The displacements that occur during the working conditions related to the acceleration and deceleration phase, respectively, are examined. To analyse such phases, two simplified functional schemes are created to easily represent the Functional Requirements (FRs) and the relative locations of parts assembled on the shaft.

During the acceleration phase both first and second wheels are pushed towards the left verse and are stopped by wheel-hub H<sub>1-2</sub> and shoulder, respectively (fig. 5). This condition occurs both when the first and second gear are engaged.

The deceleration phase could be characterized with the engaged gear. It is possible to carry out two simplified schemes; one is related to the condition of first gear engaged (fig. 6) and one related to the condition of second gear engaged (fig. 7). In the first condition the gear wheel 2 is pushed towards the left verse and wheel 1 is pushed rightward because it is still driven. In this condition the wheel-hub is not stressed along the axial direction.

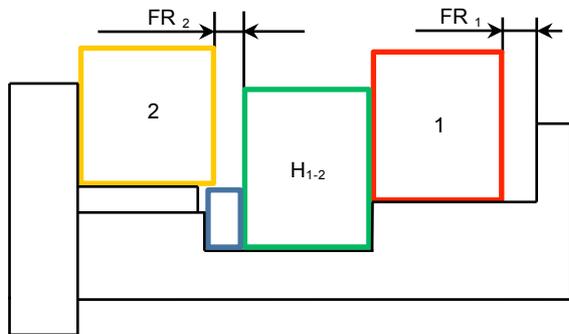


Fig. 5 Simplified functional scheme for the acceleration phase.

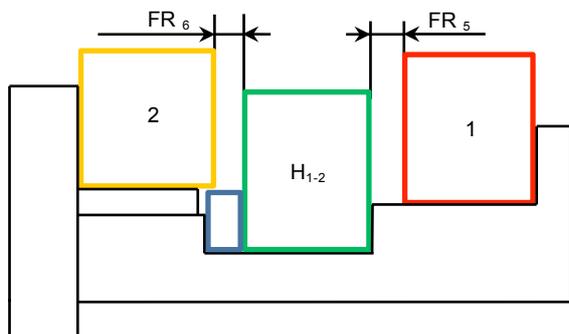


Fig. 6 Simplified functional scheme for the deceleration phase when the first gear is engaged.

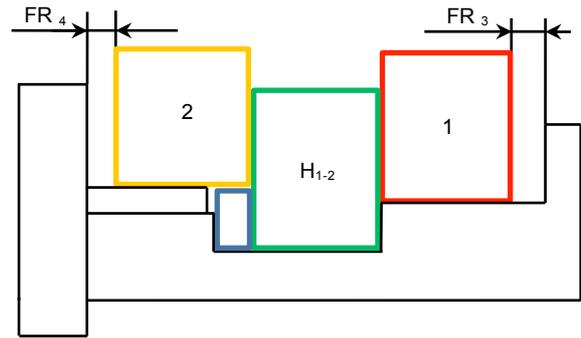


Fig. 7 Simplified functional scheme for the deceleration phase when the second gear is engaged.

Instead, when the second gear is engaged (fig. 7), the wheel 1 moves leftward while the wheel 2 moves rightward. Both are stopped by wheel-hub.

The proposed approach in [7] is applied on Group 2 composed by the first and the second gear wheels, the wheel-hub 1-2 and the shaft surfaces depicted in fig. 4. A similar dimensioning scheme is applied to other two functional groups to accomplish the axial dimensioning of the whole shaft. The simple configuration depicted in fig. 4 and characterized by three surfaces allows to fix three dimensioning schemes.

The three dimensioning schemes and the corresponding Total Information Contents are compared. The three dimensioning schemes are:

- the (proposed) functional dimensioning scheme A,
- the parallel dimensioning scheme B,
- the parallel dimensioning scheme C.

The functional dimensioning scheme A (fig. 8), is proposed according to [7]. The parallel dimension scheme is defined according to the standards for dimensioning [17] (fig. 9 and fig. 10). In particular, for the dimensioning scheme B (fig. 9) the surface S1 (fig. 4) is the reference surface while the remaining surfaces as functional; for the dimensioning scheme C (fig. 10) the surface S3 (fig. 4) is the reference surface, while the remaining surfaces are functional.

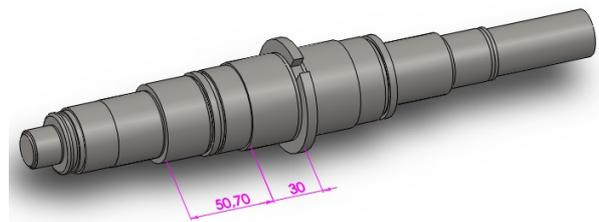


Fig. 8 Functional dimensioning scheme A.

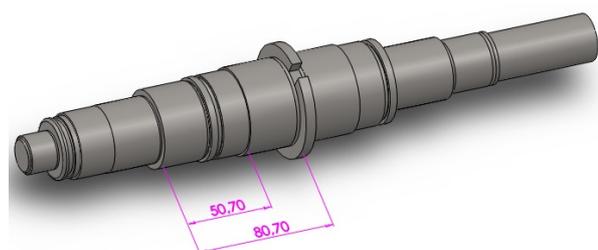


Fig. 9 Functional dimensioning scheme B.

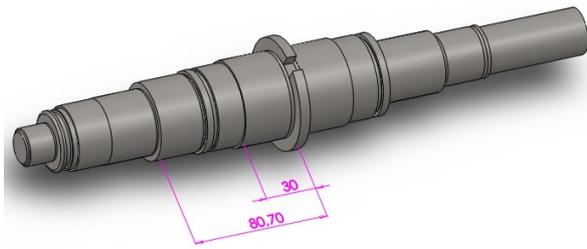


Fig. 10 Functional dimensioning scheme C.

### 5 Optimal dimensioning scheme

Tab. 1, sums up the values of *Total Information Contents* and it reveals the minimum value for the functional dimensioning scheme A. Besides, the second term of total *Information Content*,  $I_{Fr}$  that represents a loss of Information Content, is equal to zero for the dimensioning scheme A. This means that there is no a *loss of information* when the A dimensioning scheme is applied to the whole set of FRs. In other words, when the A scheme is extensively used for the whole set of FRs there are no growing effects on the values of total FRs variability.

Information Content	Dimensional Schemes (DS)		
	A	B	C
$I_D$	320,77	321,15	332,33
$I_{Fr}$	0,00	45,98	44,38
$I_{tot}$	320,77	367,13	376,71

Tab. 1 Values of total Information Contents related to different dimensioning schemes.

The relationship among FRs and nominal dimensions of parts, which particularizes eq. 1, is the one in eq. 6:

$$\begin{matrix} FR1 \\ FR2 \\ FR3 \\ FR4 \\ FR5 \\ FR6 \end{matrix} = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 & -1 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 & 0 & -1 & -1 \\ 0 & 0 & 1 & -1 & 0 & 0 & -1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & -1 & -1 \\ 0 & 0 & 0 & 0 & 0 & 1 & -1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & -1 & 0 & -1 & -1 \end{bmatrix} * \begin{matrix} D1 \\ D2 \\ D3 \\ D4 \\ D5 \\ D6 \\ d7 \\ d8 \\ d9 \end{matrix} \quad (6)$$

In particular, the vector of design parameters (eq. 6) is composed both by the dimensions of the shaft ( $D_i$ ) and the dimension of wheels and hub ( $d_i$ ).

- dimensions  $d_i$ , related to gear wheels and hub, does not change within each of the three dimensioning schemes;
- dimensions  $D_i$  are generally different within the three dimensioning schemes;  $D1$  and  $D2$  are related to the dimensioning scheme A;  $D3$  and  $D4$  are related to the dimensioning scheme B in which  $D4=D2$ ; dimensions  $D5$  and  $D6$  are related to the dimensioning scheme C in which  $D6=D1$ ;
- the three sub-matrixes highlighted in eq. 6 are related to the scheme A, B and C, respectively;
- only the sub-matrix on top-left is diagonal so only the dimensioning scheme A in according to [15] is functional.

According to the methodology in [7] and to the results in [12] and [15] it is possible to fix the scheme A as optimal scheme. In fact it is the scheme with the *diagonal chain matrix* and it is determines the minimum value of *Information Content*.

#### 5.1 Definition of the optimal dimensioning scheme over the three working conditions

All FRs were assessed using the worst-case approach. The values of FRs were evaluated assuming a Js class for tolerance and an IT grade (ISO grade) equal to IT8.

The nominal values associated to FRs are the same for each dimensional scheme, while the extreme values and the deviations are different. Using a worst-case approach the values of axial clearances for the first and the second gear wheels (wheel 1 and wheel 2) have been evaluated (tab. 2). These axial clearances determine the maximum shift lever movement.

The nominal values and the tolerances for FR1-FR3-FR5 and for FR2-FR4-FR5 are the same (tab. 2 and tab. 3). The maximum values in worst-case analysis (tab. 3) are in column A. This evidence confirms that scheme A is the optimal one.

Functional Requirements		Dimensional Schemes (DS)		
		A	B	C
FR 1	Nominal	0,100	0,100	0,100
	Max	0,143	0,176	0,143
	Min	0,058	0,024	0,058
	Tolerance	0,085	0,152	0,085
FR 2	Nominal	0,200	0,200	0,200
	Max	0,276	0,276	0,296
	Min	0,124	0,124	0,104
	Tolerance	0,151	0,151	0,192
FR 3	Nominal	0,100	0,100	0,100
	Max	0,143	0,176	0,143
	Min	0,058	0,024	0,058
	Tolerance	0,085	0,152	0,085
FR 4	Nominal	0,200	0,200	0,200
	Max	0,276	0,276	0,296
	Min	0,124	0,124	0,104
	Tolerance	0,151	0,151	0,192
FR 5	Nominal	0,100	0,100	0,100
	Max	0,143	0,176	0,143
	Min	0,058	0,024	0,058
	Tolerance	0,085	0,152	0,085
FR 6	Nominal	0,200	0,200	0,200
	Max	0,276	0,276	0,296
	Min	0,124	0,124	0,104
	Tolerance	0,151	0,151	0,192

Tab. 2 Dimensional values associated to FRs and evaluated for each dimensioning schemes

IT 8	Dimensional Schemes (DS)		
Worst-case	A	B	C
FR 1	2,720	1,973	2,720
FR 2	1,892	1,892	1,583
FR 3	2,720	1,973	2,720
FR 4	1,892	1,892	1,583
FR 5	2,720	1,973	2,720
FR 6	1,892	1,892	1,583

Tab. 3 Worst-case analysis IT8.

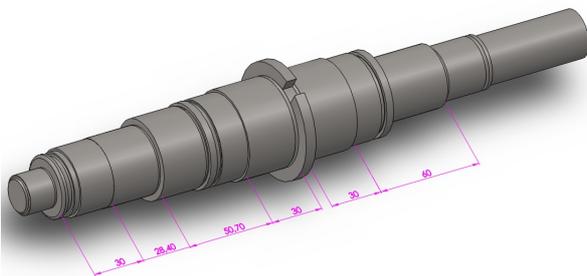
Tab. 4 shows the increases of tolerance values, evaluated for each FRs, respect to minimum values. In particular:

- the first column contains the minimum value of tolerance associated to FRi, i.e.  $\Delta FR_{J_{min}}$  in eq. 4. The minimum value is evaluated taking into account the minimum value of tolerance for each FR (see tab. 2).
- the columns A, B and C, show the difference,  $\Delta FR_{m,j} - \Delta FR_{J_{min}}$ , stated in eq. 4.
- The dimensional scheme A has for each FRs a zero value for  $\Delta FR_{m,j} - \Delta FR_{J_{min}}$

Minimum value of tolerance for each FR		A	B	C
0,085	FR 1	0,000	0,067	0,000
0,151	FR 2	0,000	0,000	0,041
0,151	FR 3	0,000	0,000	0,041
0,085	FR 4	0,000	0,067	0,000
0,151	FR 5	0,000	0,000	0,041
0,085	FR 6	0,000	0,067	0,000

**Tab. 4 Differences between tolerance values, evaluated for each dimensioning scheme, and the minimum value related to each FR**

The tolerance values obtained with the dimensional scheme A are equal to the minimum tolerance value for each FRs; in particular, for FR1, FR4 and FR5 the scheme A is better (it determines smaller tolerances) than scheme B, and equal to C scheme C; while for FR2, FR3 and FR6 the scheme A is better than C scheme and equal to B scheme. This result confirms that scheme A is the optimal dimensioning scheme for the transmission shaft. The non-null values in the columns related to B and C schemes, represent the increase of FRs tolerances due to the use of dimensioning schemes different from A. In the present study the FRs values represent gaps between parts. The tolerance values of gaps (axial clearances) are the maximum ones for the first and second gears wheels. These axial clearances determine the maximum *shift lever movement*, when the vehicle moves from the acceleration to the deceleration phase or vice-versa. The procedure applied to the Group 2 was also applied to the other functional groups in order to obtain the whole shaft dimensioning scheme, depicted in fig. 11.



**Fig. 11 Final functional dimensioning for the transmission shaft.**

## 6 Statistical analysis

The dimensioning schemes are also evaluated by using a statistical approach. A process capability index is evaluated for each dimensioning schemes, and for each FRs, respectively (tab. 5)

IT 8	CP		
	Dimensioning scheme A	Dimensioning scheme B	Dimensioning scheme C
FR 1	3,24	2,67	3, 25
FR 2	2,26	2,26	2,03
FR 3	3,24	2,67	3, 25
FR 4	2,26	2, 26	2, 03
FR 5	3,24	2,67	3, 25
FR 6	2,26	2, 26	2, 03

**Tab. 5 Process capability analysis: IT grade = IT 8.**

The values of FRs were evaluated assuming a Js class for tolerances and an ISO grade equal to IT8. CP index, by using the Eq. 7, is evaluated.

$$CP = \frac{USL - LSL}{6\sigma} \tag{7}$$

For each FRs, the nominal value contained in tab. 2, is used to evaluate the specification limits:

- Nominal(FRs) + 0.1 is the upper specification limit (USL)
- Nominal(FRs) - 0.1 is the lower specification limit (LSL).

CPK is not evaluated because CP and CPK are coincident. The Root Sum Square (RSS) method is used to evaluate the standard deviation of the process (Eq. 8).

$$\sigma_{Tot} = \sqrt{\sum_{i=1..n} \alpha_i^2 * \sigma_i^2} \tag{8}$$

$\alpha_i = \frac{\partial f}{\partial x_i}$  is the sensitive factor.

The single standard deviation is evaluated by assuming a natural tolerance ( $6\sigma$ ) for the input tolerance. The i-th standard deviation related to i-th tolerance zone is evaluated by using Eq. 9

$$\sigma_i = \frac{Max.size_i - Min.size_i}{6} \tag{9}$$

Due to the high values of CP ( $\gg 1$ ), the analysis was performed also with an IT9 grade, in order to reduce manufacturing costs. The results are shown in tab. 6.

The higher values corresponding to scheme A confirm that it is the optimal dimensioning scheme.

IT 9	CP		
	Dimensioning scheme A	Dimensioning scheme B	Dimensioning scheme C
FR 1	2,72	1,97	2,72
FR 2	1,89	1,89	1,58
FR 3	2,72	1,97	2,72
FR 4	1,89	1,89	1,58
FR 5	2,72	1,97	2,72
FR 6	1,89	1,89	1,58

**Tab. 6 Process capability analysis: IT grade = IT 9**

## 7 Conclusion

The functional approach to optimal dimensioning of automotive transmission shafts drives the designer in identifying the optimal dimensioning scheme of parts in a mechanical device, as a gearbox shaft, characterized by a mix of functionalities i.e. different positioning of the parts in the assembly. So, the approach uses an appropriate index to evaluate the Information Content for each dimensioning scheme and it addresses the optimal dimensioning scheme by minimizing the index. By associating the characteristics of parts assembled on the shaft, to the corresponding functional surfaces and, finally, to the related dimensioning schemes, three similar functional conditions are identified on the shaft. Therefore, the analysis of possible dimensioning schemes is significantly simplified and, then, by accomplishing the optimal dimensioning of a single subassembly it is possible to extend the approach and the related dimensioning to the whole shaft. The usefulness of the approach is tested by evaluating the axial clearances of different parts assembled on the automotive shaft, for several dimensioning schemes. The results are compared and a proposed functional dimensioning scheme is accepted as the optimal scheme.

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