QUALITATIVE DESIGN OF COMPACT TRANSMISSION MECHANISMS WITH STANDARD COMPONENTS

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Abstract :

This paper presents a new synthesis method for multi−stage transmission mechanisms. Standard components and standard orientations of the components in space are considered. The method comprises three phases called Exploration, Elimination and Sorting. For Elimination phase, several design rules are presented. For Sorting phase, four new performance functions connected to the compactness notion are described. The method has particular abilities for reducing the combination space and pointing out compact mechanisms in a qualitative way. A final example validates the method.

Key words : mechanism synthesis, qualitative design, speed reducer, standard component, compact.

1 Introduction

If we consider the Computer Aided Design market today, a great number of CAD softwares can be listed and most of them have similar purposes and also a similar structure. The majors actors in the world of mechanical design are made of a collection of software modules, each of them being dedicated to a particular design activity. The vast majority of softwares feature et least a *"design" module* for drawing 3D mechanical parts, a *drafting module* for making 2D plans, an *assembly module* for creating mechanisms from several parts, an *engineering module* for performing kinematical analysis as well as resistance calculations and generally also a *manufacturing module* for creating parts with numerical command machines. This collection of modules offers a wide covering of most of all the activities of a mechanical designer.

Another strong tendency of the market is the recent arrival of softwares that are ordinarily called "Wizards" or "Assistants". These programs try to summarize the know−how of a specialist in a restrained area of design, thus enabling the user to create complex mechanical parts by themselves without any deep knowledge of the technical problems. Several products already exist in the domains of mold design and stamped parts.

However, as far as we know, one could hardly find any wizard or dedicated module for preliminary sketching and design of machines. The initial task of the designer still remains to search good ideas by himself, to imagine and compare various general machine architectures and then to determine, often manually, which of them will be worth digging with a CAD software.

In this paper, we present what could be a "preliminary design wizard" for transmission mechanisms. We presented in a previous publication a first original method for designing such mechanisms [1]. This method was a general one and already permited to quickly explore a considerable domain of feasible solutions. The evaluation of the pertinence of solutions could be done in a multi−criterion way or thanks to a fuzzy logic method [2]. The drawback

of this general approach was to be insufficiently precise in the mechanism definition and to propose a great number of solutions without always being capable of differentiating them in order to choose the best one. In this work, we will consider a different design method based on a standard component model, with standard geometric orientations. We will show that even more qualitative information can be extracted from such a model.

Several other works in the domain of transmission mechanism synthesis can be found. Chakrabarti and Bligh present a method for designing force and motion transmission and transformation mechanisms [3,4,5]. The mechanisms to be designed are described by their input and output, with multiple I/O if necessary, and are made of combinations of elementary mechanical modules. The method uses design rules for producing graph structures of solutions. Sense and orientation constraints are also considered with « orthogonality restrictions », which is also a principle of the present article. Kota and Chiou propose a synthesis method of compound mechanisms based on a qualitative matrix representation of elementary mechanisms considered as building blocks [6]. Both of these methods are interesting but do not focus on the geometrical point of view of compactness.

These were some synthesis methods. On the other side, Joskowicz and Sacks describe qualitative tools for performing kinematic analysis of mechanisms, particularly of fixed−axes mechanisms including gear boxes and transmissions [7,8]. Inputs are mechanism shape and initial configuration. Outputs are a region diagram showing mechanism behaviour from its configuration space. Forbus worked on qualitative spatial reasoning for understanding complex mechanical systems [9]. Qualitative kinematics and dynamics are taken care of and the example of a planar geared clock mechanism is treated.

It seemed interesting to us to try to mix, in a same method and software, a synthesis phase in the spirit of [3−6] with a qualitative analysis for getting a better characterization of proposed solutions. Our qualitative analysis is not directly comparable to [7−9] because the considered class of mechanisms does not exactly needs the same tools, but the principle of understanding a mechanism in a global way without dimensions and precise values stays the same.

2 Principles of the new method

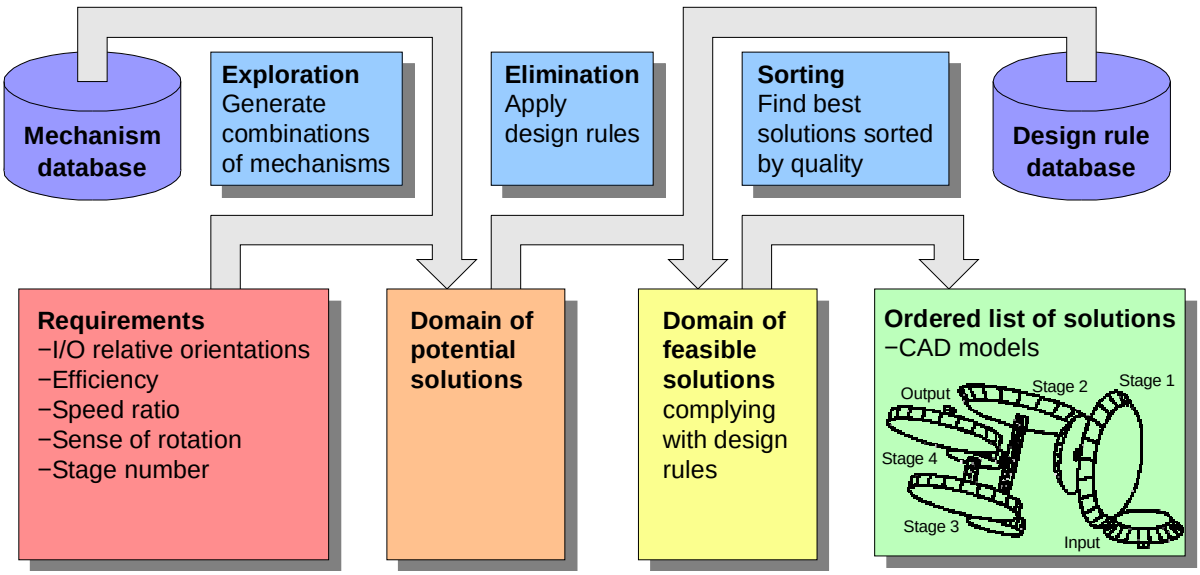

Fig. 1 : *Architecture of the original method*

We consider the problem of designing transmission mechanisms such as speed reducers, complying with specifications like speed ratio or minimum efficiency. Of course, some specific high ratio compact original mechanisms exist, as pointed out in [10] (i.e. harmonic, Cyclo or Rota drives). However, most of common speed reducers have a chain structure in

which several stages are combined one after the other. This chain structure offers a great flexibility in terms of geometric shape and high ratio, as well as a competitive price and good scalability, but it also seriously complicates the design of this type of mechanisms. Which stage to chose ? In what order ? Where should it be located in space ? These are the questions the designer should be able to answer in the initial design phase, being conscious that what will be answered here might have critical consequences on the final design.

It is for solving this awkward problem that was developed our original qualitative design method. The general structure of the method can be seen in Fig. 1 and keeps unchanged for the new method. It is made of three main phases called Exploration, Elimination and Sorting, that will be described with more details in sections 6 to 8 with their new enhancements.

The method starts from the initial requirements of the user, which are extremely reduced :

- The unique geometrical specification is relative orientation of input and output shafts. They can be parallel on opposite sides, parallel on the same side, orthogonal or at a specific angle.
- Then come mechanical pieces of information, like the global minimum efficiency, global speed ratio or output sense of rotation for a positive input rotation
- The last requirement is about the maximum number of stages the global mechanism should be made of. Of course, simpler mechanisms will also be listed.

With this specification sheet, the original method was able to construct “qualitative solutions”, that is to say to represent the topology of the mechanism, its principal parts and their relative ordering. However, such a “qualitative” model should not be understood as the final model, because all the parts still have the potentiality to change dimensions and spatial orientation. The only information included in the model is connectivity between parts, and the CAD representation of the model should be understood as a “rubber model”, likely to be warped according to later design constraints. For giving an idea of the great changes that might affect a qualitative model, figure 2 shows three possible aspects of a given model. In Fig. 2b, various part dimensions were modified relatively to Fig. 2a. In Fig. 2c, angles of bevel gear stages are no more right and angles of consecutive stages around their common axis were also changed, which upsets completely the mechanism geometry.

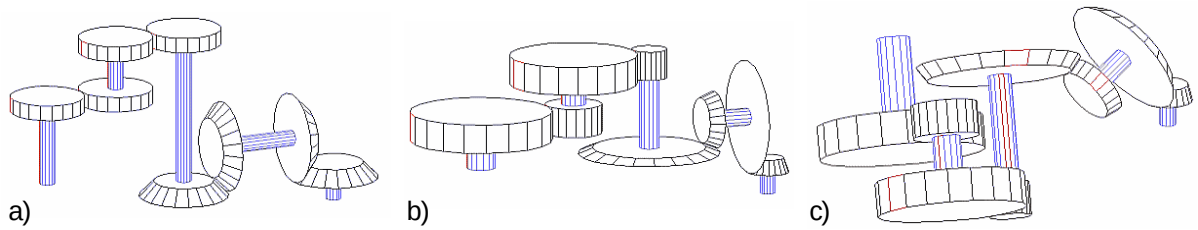

Fig. 2 : *A qualitative model, as understood in the original method, has no fixed dimensions*

From this example, it can be seen that the original qualitative model is rather poor in information. This can also be observed in [1] at the end of the original method, when the final list of solutions is shown to the user : it frequently occurs to obtain twenty solutions ranked at the first place, among several other hundreds. This shows clearly there is no way of finding the very best mechanism out of the nineteen others because data is too vague and fuzzy.

The basic idea of this new method is to try to enhance the precision of the results given by the original synthesis method. In the new method, we will try to re−create the designer behaviour when he sketches mechanical architectures on a paper and notices interesting configurations for choosing the best one.

3 Standard orientations

If we want to evaluate the interest of a mechanism more precisely, we saw that its qualitative model should have to be enriched a little. In order to keep thinking “qualitative” and re−

produce the way a human designer thinks, a natural idea would be to constrain the model of the mechanism by assigning fixed values to all the angles.

Angles are non−dimensional parameters that greatly influence the whole design of a mechanism. Keeping in mind we want to perform a rough comparison of various architectures, the following simplification will be assumed : angles will only take round values among 0°/90°/180°/270°.

This is a strong assumption but it will permit to rapidly explore a great number of mechanism layouts. Moreover, if we consider most of industrial speed reducers, it is extremely rare to see “original” orientations of parts, because of considerations like opening the casing, fixating bearings or standardising parts. Consulting a standard catalogue will confirm this [11].

However, even with this simplification, a given mechanism can take a great number of layouts, as shown on Fig. 3. Exploring the four main angle configurations permits to have a quick view of all interesting configurations. In specific cases, it would be possible to further refine angle values with optimisation tools.

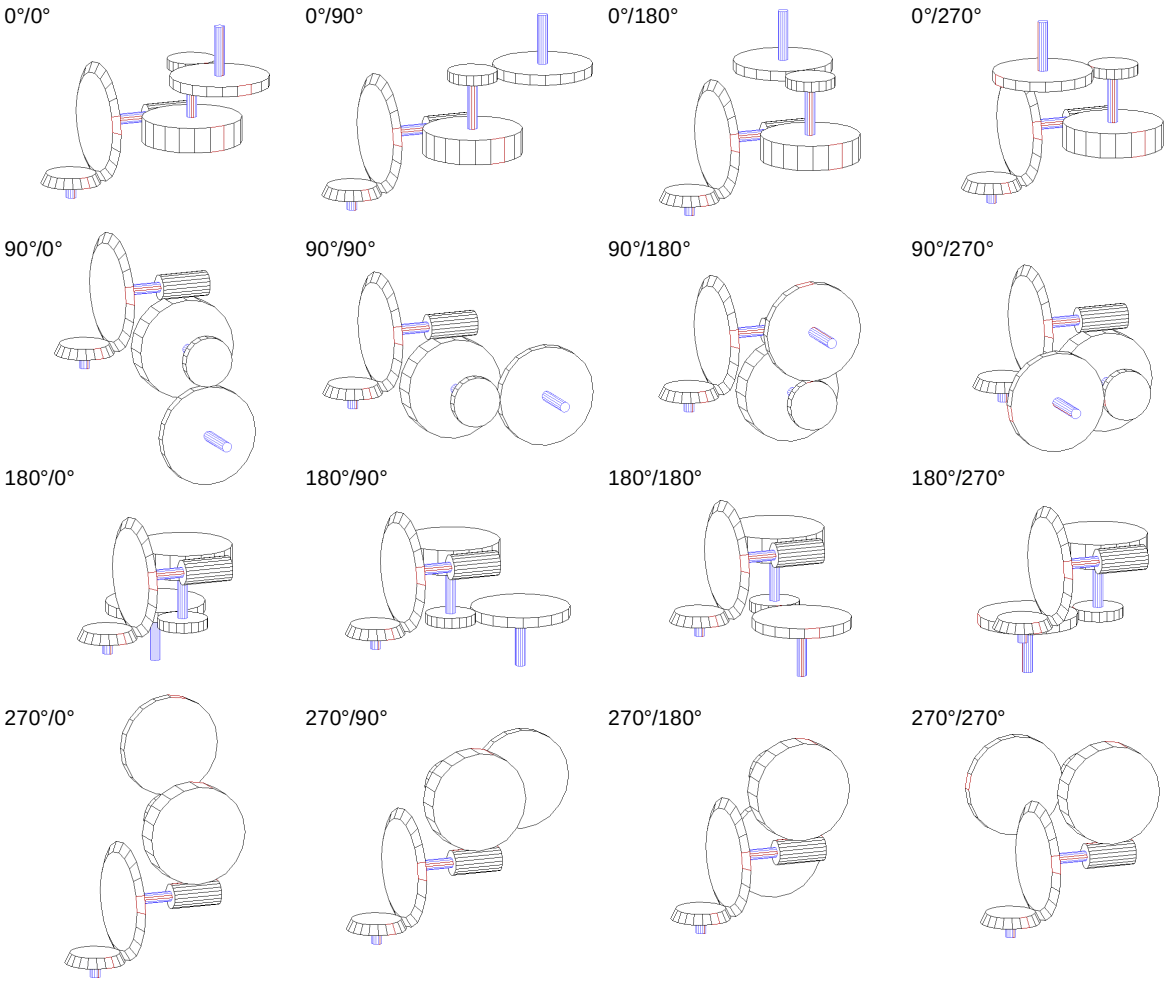

Fig. 3 : *The sixteen configurations of a three stage mechanism with angles between stages.*

4 Compactness of a mechanism

One of the best ways to differentiate several good solutions one from the other is to look at their compactness. We saw on Fig. 3 that a same mechanism can provide a great variety of layouts. Some of them are particularly flat, some are wider, others are rather cubic. When we say a mechanism is “compact”, we mean that it perfectly fits the space available for it, with no spoilt space inside or, on the contrary, big parts crossing outside (Fig. 4). If we remind the fixed angle hypothesis of section 3, we can see on Fig. 4a there is no way to make the me−

chanism stay in the available space if we try at the same time to keep the output orientation. Moreover, the general envelope of the mechanism is rather vertical instead of being horizontal as required. This layout of the mechanism proves definitely inadequate to the geometric specifications, even if the speed ratio, rotation sense and efficiency are good. This type of qualitative information on compactness is what we are trying to add into our new design method.

This naturally shows that initial specifications given to our new method will necessarily include data on the available space. Of course, data is provided in numerical form (lengths, I/O shaft coordinates, etc.) but our interpretation will try to stay qualitative and non−dimensional.

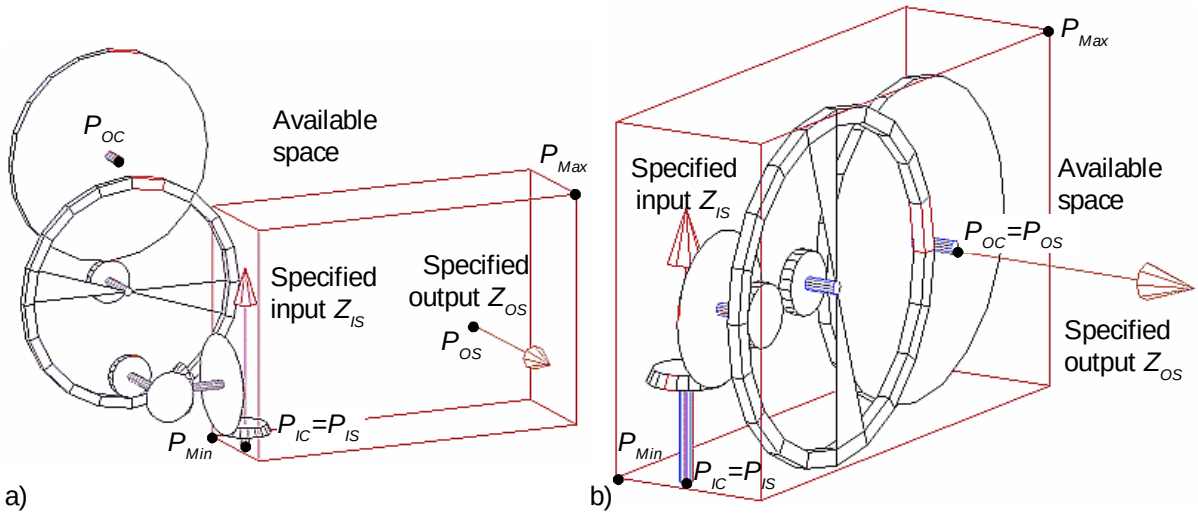

Fig. 4 : *A non−compact and a compact mechanism for a same available space.*

5 A database of standard mechanisms

Evaluating compactness will be difficult with mechanisms with no dimensions at all, even from a qualitative point of view. For this reason, we decided to work with *semi−dimensioned* elementary mechanisms. Let us consider a simple cylindrical gear for instance. In the original method, such a component did not have pre−determined diameters, so the ratio was not fixed. All what was known was that this ratio ranged from 1 (same diameters) to 8 (small pinion, big wheel). In the new method, we decide to address only mechanisms with determined mechanical parameters. Consequently, parameters like ratio or efficiency become fixed instead of being fuzzy. This is the reason why the new method is suitable for “standard components”, be−cause every single mechanism for a stage may be taken from an industrial catalogue.

However, it should be noted that even if the mechanism may be completely defined, it is not absolutely necessary. This is what we mean by *semi−dimensioned*. For instance, gear tooth width may be kept undefined for the moment, because we are interested only in the transmission function, that is to say on speed ratio and diameters. Gear tooth width is less important for overall dimensions and is tightly linked to transmittable torque. It will be determined later in the design process but certainly not in the qualitative model.

We now introduce the database of semi−dimensioned elementary components that will be used in the new method (Fig. 5). Database includes twenty representative mechanisms for demonstration purpose but might be extended to a full catalogue if necessary. In order to be comparable, 3D models were all based on a reference dimension *R*, which is the radius of all the pinions. Here are specific remarks on every type of mechanism :

– **Cylindrical gears sets (1−6).** Two settings were considered : opposite shafts or shafts on the same side. This has great consequences on the global mechanism topology. Chosen ratios are 1, 2, 4. A ratio of 1 is useful for only reversing sense. A ratio of 8 is possible but the gear set is quite big. With ratio 4, the biggest dimension of the gear set is already *10R*.
– **Internal cylindrical gears sets (7−10).** Two shaft settings are available again. It should be

noted that in the case of shafts on the same side, the pinion may interfere with output shaft. This is why the ratio 2 was not chosen. Chosen ratios are 3 or 5. Internal gear sets are more compact that external ones, for a given ratio. For example, an internal gear set of ratio 3 has a *6R* biggest dimension while an external gear set of the same dimension only reduces 2 times. Similarly, internal gears of ratio 5 are as big as external gears of ratio 4.

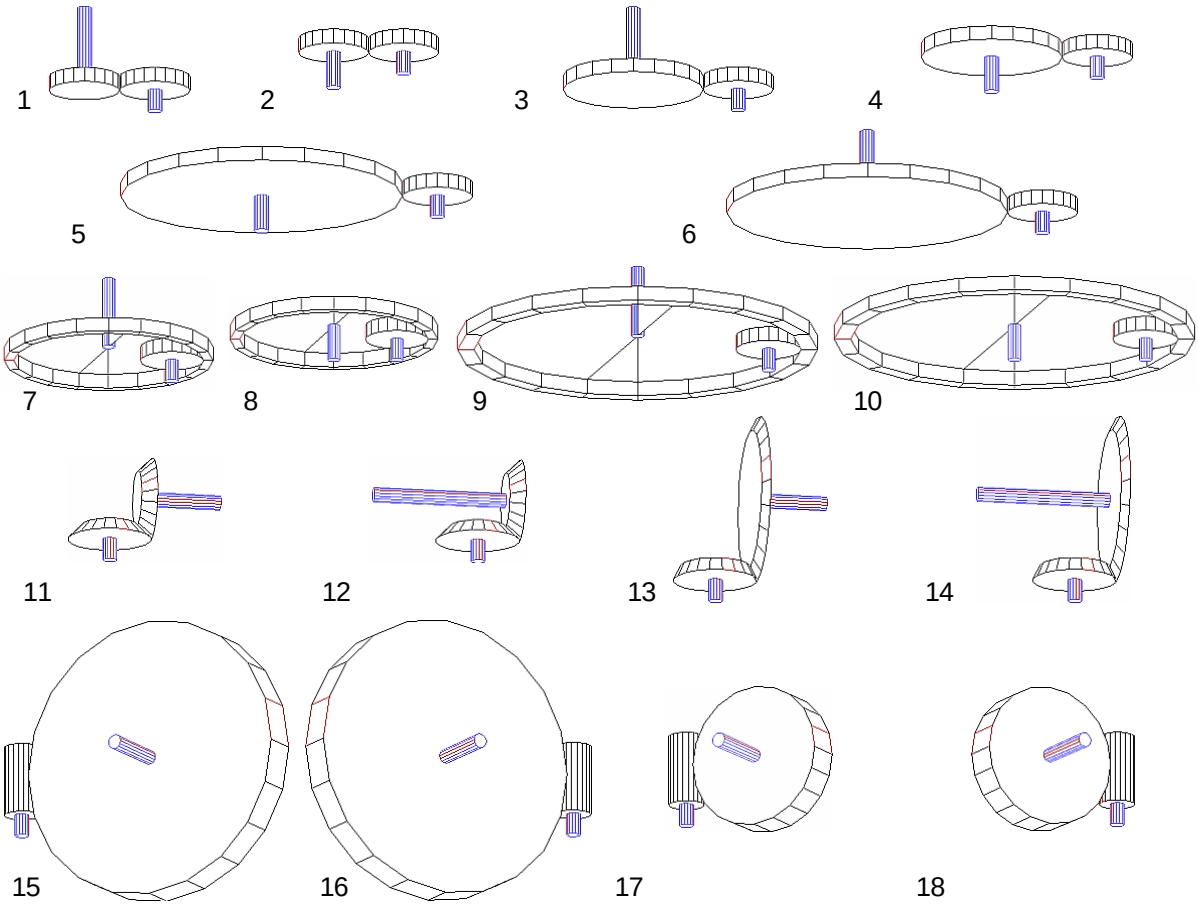

Fig. 5 : *Database of twenty semi−dimensioned components represented at the same scale.*

- **Bevel gears sets (11−14).** The second shaft setting (12 and 14) reverses sense of rotation. With a 1 ratio, the stage has only the function of changing direction, while it has also a reduction function with ratio 2. Ratios could be bigger than 2 with bigger wheels. However, as they are more expensive than cylindrical wheels, catalogues generally offer less choice of big ratios than with cylindrical gears.
- **Worm gears sets (15−20).** These stages are compact with big ratios. Two shaft settings are available. Such mechanisms have a wide ratio interval (from 10 to 500) but also a great variability in efficiency (from 90% to 30% respectively). With the original method, this type of gear set was problematic because ratios and efficiencies were too vague, so resulting mechanisms using worm gears had imprecise specifications. With the new method and standard components, it is possible to specify the exact efficiency corresponding to a given ratio. Here are the three representative chosen mechanisms :
  - 4 thread screw with 32 tooth wheel, ratio of 8, correct efficiency of 85% (15−16)
  - 1 thread screw with 16 tooth wheel, high ratio of 16, medium efficiency of 75%, ultra−compact size (17−18)
  - 1 thread screw with 32 tooth wheel, very high ratio of 32, but low efficiency of 67% (19−20, same geometry as 15−16)

  This choice permits to illustrate the two following design rules :
  - With fewer screw threads, ratio increases but efficiency decreases.

– With more wheel teeth, ratio and overall dimensions increase but efficiency decreases.

6 Exploration

This is a combinatorial exploration. It can be understood as running a special counter of configurations (Fig. 6). If the maximum number of stages is set to N_{MaxS}, the counter should have two blocks of N_{MaxS} figures. The N_{MaxS} figures on the left represent the nature of consecutive stages and are a counter in base $N_{MaxM}+1$ where N_{MaxM} is the number of mechanisms in the database. The N_{MaxS} figures on the right are a counter in base 4, with figures taking 4 distinct values : 0°, 90°, 180° or 270°.

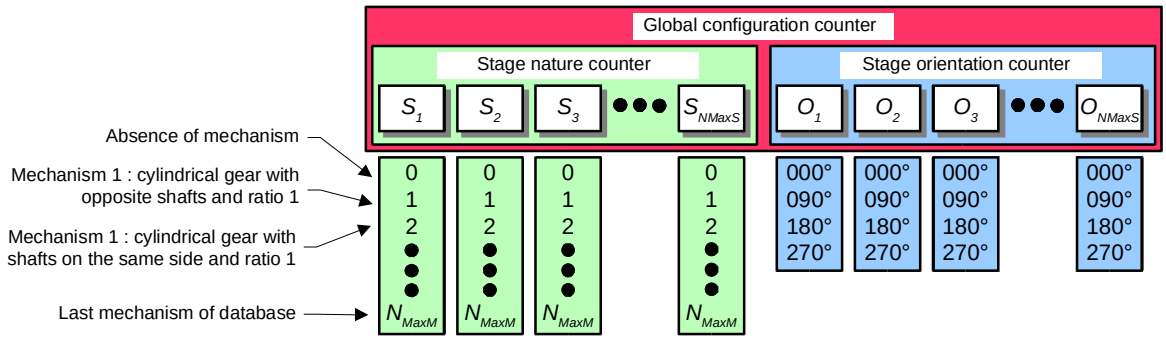

Fig. 6 : *The mechanism configuration counter.*

7 Elimination

By adding to the original method the exploration of standard orientations, we considerably increase the size of the domain of configurations to explore. So we can expect to obtain much more solutions. Fortunately, as we are now using standard components and standard orientations, we can perform a much more efficient elimination phase. In order to be considered feasible, a mechanism configuration must satisfy several constraints :

– **Limitation on the number of stages with orthogonal shafts.** As angle between global I/O shafts may only be a multiple of 90°, no need to use too many expensive orthogonal stages. This design rule limits their number to two and their position to the two first stages, because they are submitted to small torque, thus ensuring small and inexpensive parts.

– **Good efficiency** : $\eta_C \geqslant \eta_S$ with $\eta_C = \prod_{i=1}^{N_{MaxS}} \eta_i$ (1)

Relation (1) means considered configuration must have better efficiency than specified.

– **Good speed ratio** : $U_S - \Delta_U \leqslant U_C \leqslant U_S + \Delta_U$ with $U_C = \prod_{i=1}^{N_{MaxS}} U_i$ (2)

where S indices refer to specified data, C indices to global configuration characteristics and i indices to stage characteristics.

Relation (2) means considered configuration must have the specified reduction ratio U_S. Of course, depending on the ratios of standard elements in database, some global ratios are not attainable. Thus, user should also specify a tolerance Δ_U.

– **Good rotation sense** : global sense is obtained by combining senses of each stage.

– **Good absolute orientation of output shaft** : $\vec{Z}_{OC} = \vec{Z}_{OS}$ with $\vec{Z}_{OC} = \prod_{i=1}^{N_{MaxS}} \mathbb{R}_i \vec{Z}_{IC}$ (3)

with $\mathbb{R}_i$ being the rotation homogeneous matrix that transforms input axis into output axis of a given stage. This matrix was previously stored into the mechanism database for each stage. Vector $\vec{Z}_{OS}$ represents the specified output vector, as it can be seen in Fig. 4.

– **Correct relative I/O locations** : $\widehat{(\overrightarrow{P_{IC}P_{OC}}, \overrightarrow{P_{IS}P_{OS}})} < 90°$ with $P_{OC} = \prod_{i=N_{MaxS}}^{1} \mathbb{C}_i P_{IC}$ (4)

with $\mathbb{C}_i$ being the combined translation and rotation matrix that transforms input refer−

ence axes into output reference axes of a given stage. This angle is also called γ and visible on Fig. 7a. Condition (4) ensures that configurations like the one on Fig. 4a are automatically rejected because of their bad global relative locations of input and output.

8 Sorting

Feasible mechanism configurations obtained after the elimination phase are often in great number. In order to facilitate the user choice, best configurations are sorted first. Five performance functions are defined below. The first one is directly inherited from [1] :

– **Mechanism nature quality :** a first way of evaluating the interest of a mechanism is to analyse the stages it is made of. Each stage is given a mark according to three criteria : power transmitting ability, fabrication cost and mounting cost. A weighted mean of each stage marks permits to obtain a global mark F_M for the mechanism. Weights are user configurable.

Four new performance functions are defined in the new method for extracting the configurations that match various geometric aspects of initial requirements. These functions will help measure the compactness of a given configuration.

– **Overall proportion quality :** $F_P = \left|\frac{a}{A} - \frac{b}{B}\right| + \left|\frac{b}{B} - \frac{c}{C}\right|$ with notations on Fig. 7a (5)

For calculating *a*, *b* and *c*, we have to get point $P_{Min\ i}$ and $P_{Max\ i}$ for each stage *i* (Fig. 7b). We obtain each of those diagonal points by multiplying the initial diagonal points of the standard component, stored in the mechanism database, by the transformation matrix $\prod_{j=i}^{1} \mathbb{C}_j$ for every stage *i*.

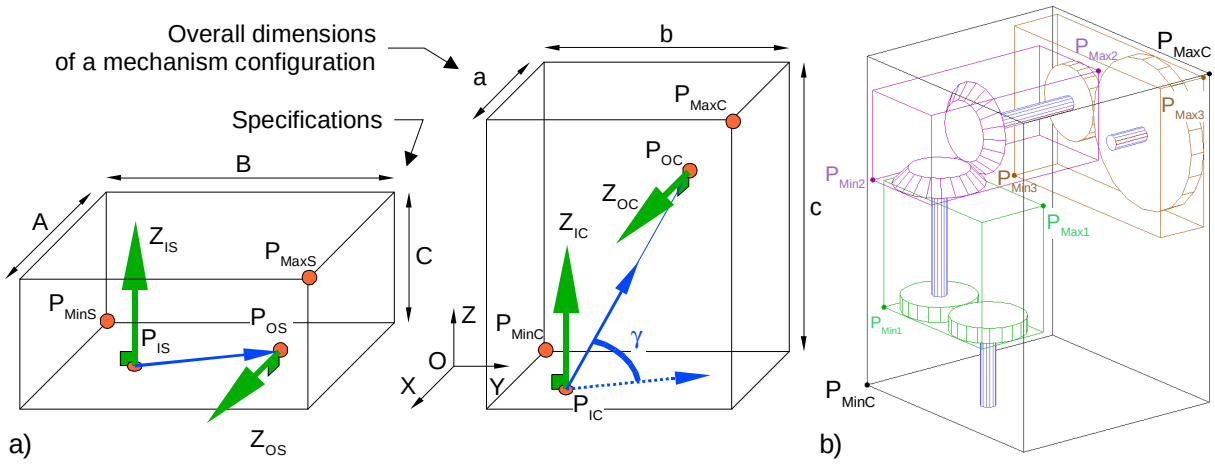

Fig. 7 : *Matching the geometric initial requirements.*

As we used exclusively angles that are multiples of 90°, it seems natural to evaluate the external envelope of a mechanism configuration by its bounding box. Relation (5) gives a non−dimensional value varying from 0 (for perfect proportion between specifications and mechanism envelope) to infinity (for an infinite extension along one or several axes).

– **Input location quality :** $F_I = (\ |NX_{IC} - NX_{IS}| + |NY_{IC} - NY_{IS}| + |NZ_{IC} - NZ_{IS}|\ )/3$ (6)

with $NX_{IC} = \frac{X_{IC} - X_{MinC}}{X_{MaxC} - X_{MinC}}$ and $NX_{IS} = \frac{X_{IS} - X_{MinC}}{X_{MaxC} - X_{MinC}}$ and same definitions along *Y* and *Z*.

NX_{IC} represents the non−dimensional position (in %) along *X* of P_{IC} on the box. So F_I measures the non−dimensional distance between the specified input position and the current input position. It ranges from 0 (for a perfect coincidence) to 1 (for diagonally opposed points). Function F_I permits to measure distance between a vague specification such as "input located just in the corner" and a qualitative solution with, for example, "the mechanism input near the opposite side".

– **Output location quality :** a quality function F_O is similarly defined for output.

– **Relative I/O location quality :** $F_{IO} = 1 - \cos(\gamma)$ (7)

Angle γ is defined on Fig. 7. As γ is smaller than 90° because of elimination relation (4), F_{IO} is between 0 (perfect similarity between relative I/O locations) and 1 (configuration very far from specifications).

Mechanism configurations are sorted first by F_M, then F_P, then $F_I+F_O+F_{IO}$.

9 Example

Not let us solve the problem of designing a speed reducer with orthogonal I/O shafts, minimal efficiency of 90%, speed ratio around 47, reversing sense and no more than 4 stages. We also give the following : $P_{MinS}=(100,100,100)$ $P_{IS}=(200,150,100)$ $Z_{IS}=(0,0,1)$
$P_{MaxS}=(350,900,650)$ $P_{OS}=(350,650,350)$ $Z_{OS}=(1,0,0)$

With $\Delta_U=0$, we find no solution at all, which is normal because a ratio of 47 cannot be obtained with the components in the database. With $\Delta_U=1$, we find 5723 solutions. As we swept a space of 10 240 000 configurations, the method reduced the space size by a factor 2000, which is good. Three interesting solutions can be seen in Fig. 8b−8d. They look very close to the initial geometric specifications (Fig. 8a). On the contrary, one of the worst solutions is given in Fig. 8e.

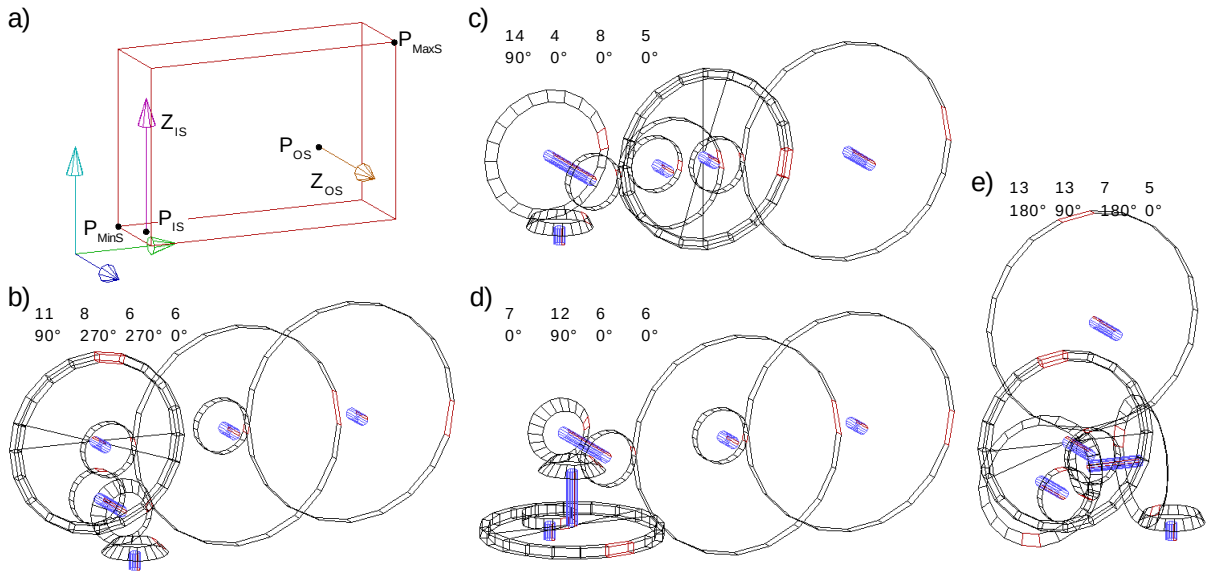

Fig. 8 : The g*eometric specifications, three good solutions and a less good one.*

The whole calculation was performed in less than 5 seconds on a Pentium III microprocessor running at 650 Mhz. With 5 stage combinations, calculations never last longer than 5 minutes. Most of the time is spent in Exploration and Elimination phases. Sorting is never very long because of the excellent performances of the sorting method, a recursive method called QSORT and provided by standard Ansi C language. This method has an average running time of $N.log(N)$ with N being the size of the ensemble to sort. In our case, sorting 10^6 elements lasted about 10 seconds.

Another tip for cutting down execution time is to evaluate faster elimination rules first, thus avoiding to evaluate longer ones for most of combinations. For instance, switching absolute orientation rule from the first to the last place reduced computation time from 45 to 4 seconds on a same problem.

10 Conclusion

We presented a new method for designing transmission mechanisms with standard components and standard orientations. This method is highly improved and performs significantly better than the original method presented in [1] for various reasons :

- using standard components permits to store more precise and realistic data in the mechanism database, particularly for efficiency and ratio
- solutions at the output of the new method are better defined and contain more qualitative information, with realistic diameter proportions and exact 3D part orientations
- thanks to ne notion of compactness, it is now easier to point out good mechanisms among a great number of solutions that previously had the same rank.

In the future, improvement have still to be done in the expression of elimination constraints and sorting functions for refining the final population of solutions. Algorithm enhancements should also be considered for faster computations with large component databases.

Nevertheless, the method proved to be an efficient tool for suggesting initial mechanism architectures to a designer. It permits to divide by several thousands the size of the initial combination space and forces the designer to exhaustively consider ALL the available solutions. This is one more step toward computer aided design of transmission mechanisms.

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