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Decision support tool for clearance management in a conceptual design context

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Abstract

It is well known that manufactured over-constrained mechanical systems cannot operate without clearance between parts. If the clearance value is too small, parts will not fit together. Conversely, if the clearance value is too large, parts float relative to each other and the movement is imperfect. This paper presents a tool providing decision support during the conceptual design of over-constrained mechanisms. A new key characteristic of the mechanical system, named F , is defined. F quantifies the floating phenomenon of the mechanical assembly. The smaller it is, the smaller is the free movement of parts around their nominal positions. Optimizations are used to compute the minimal clearance value to guarantee the mechanism assemblability and also the corresponding F value. These values are obtained with respect to a quality level of the manufacturing process. Then, the proposed tool plots diagrams. It shows areas where the design is possible and highlights inconsistencies. We consider that it is very useful to assist users during the conceptual design step. A case study including a planar mechanism made of 6 parts and 8 joints illustrates the tool.

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Keywords: Clearance; Deviation; Over-constrained mechanism; Conceptual design; Floating phenomenon

1. Introduction

An over-constrained mechanical assembly can only be assembled or mobile if shapes and dimensions of its components meet strict geometrical conditions. Due to the manufacturing process, those conditions cannot be exactly respected, as shown by Meng et al. [1]. To deal with this problem, clearances are generally added in the system's joints to allow tiny free movement of manufactured parts around their nominal positions. Unfortunately, consequences of this solution are not always under control. On one hand, if joint clearances are too limited with respect to manufacturing capability, the free movement of parts is too small and some manufactured parts cannot be assembled with the others. Therefore, some products will be rejected. On the other hand, if joint clearances are too large with respect to manufacturing capability, the free movement of parts is important and their actual positions can be far away of their nominal positions, as

explained by Ting et al. in [2]. This generates vibrations which can affect the performances of the manufactured products. This tiny motion phenomenon is named floating phenomenon in the following.

The goal of the present work is to provide the mechanical designer with a tool to anticipate the said floating phenomenon in the assembly at conceptual design step. This tool is a diagram highlighting the relationships between joint clearances and floating phenomenon for a given manufacturing capability. This diagram is drawn at an early stage of design process, and can be used as a decision-making tool for designers.

Recent research is focused on statistical analysis during detailed design phase. For instance, in [3], a global sensitivity analysis of a mechanical system using the Sobol's method is made by Idriss et al. In [4], Homri et al. compute the probability of failure of over-constrained assemblies using quantifier based mathematical formulation. In [5], Ballu et al.

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study mechanical assemblies with clearances. Authors propose a method to compute the geometrical reliability of a product. It is applied on a completely defined mechanism composed of six parts. In several studies, mechanical motions are also taken into account as in [6] and [7]. Walter et al. study time-depending functional key characteristics of a crank mechanism, while X. Huang et al. evaluate the rotation deviation of the output link of a four-bar mechanism.

Goetz et al. in [8] propose an interesting approach to first tolerance evaluations when shapes of parts are not completely defined. Nevertheless, some effective tool is still needed for designers to predict if the manufactured products will perform as required. As demonstrated by Ebro et al. in [9], this prediction is helpful during the early stages of product development.

In [10], [11] and [12] Rameau et al. and Mabire et al. present a geometrical model designed to predict, at early phase of development, the size of the minimum clearance needed to guarantee the assemblability of over-constrained mechanical assemblies in the context of mass production. Thanks to this proposal, mechanical assemblies can be characterized, at early design step, by a quantified property named clearance (noted J). If this quantity is too small, large number of manufactured parts may not fit together.

In addition to clearance J , a second mechanical assembly property is defined, the so called deviation. The deviation value is a distance between the overall shape of a manufactured product and its theoretical nominal shape. The deviation value vanishes when the manufactured product and the nominal product have the same overall shape. The more different are the two shapes, the greater the deviation value is. Due to the floating phenomenon, a manufactured assembly features several configurations, each one corresponding to a particular overall shape. One of these shapes is the most distant: when the deviation is maximal. Another one is the closest: when the deviation is minimal. The difference between these two extreme values is called the floating.

Section 2 presents an algebraic representation of a non-nominal linkages. Section 3 defines four global key characteristics of non-nominal linkages. The dependency between these quantities is captured through two diagrams, as explained in section 4. Section 5 implements the solution on the assembly test case.

2. Algebraic representation of non-nominal assembly

A mechanical assembly is made of rigid bodies connected together by mechanical joints. Each rigid body, also called mechanical part, is manufactured by using production machines through a well-defined manufacturing process. In real life, because of unavoidable machining errors, it is impossible to get two instances of a given part featuring exactly the same dimensions. Consequently, mounting an over constrained assembly is a priori impossible and some extra work is needed. Firstly, the pairing solution is to include in the assembly one or more adjustable parts. A second solution is to allow non rigid parts that can be slightly deformed at the mounting step in order to compensate dimensional errors. The third solution is to introduce clearance in mechanical joints.

First solution is not efficient for large scale production because mounting time is substantially increased. The second solution induces parts internal strength, which can lead to poor fatigue resistance. The third solution allows tiny motions of parts in the neighborhood of their respective nominal positions, which, in turn, may induce unwanted vibrations. This paper deals with the third solution.

2.1. Closure equation of nominal assembly

Classically, a rigid assembly is modeled through a so-called closure equation. This equation is based on the closure mapping noted $f: \mathbf{U} \times \mathbf{P} \rightarrow \mathbf{E}$ where \mathbf{U} is the space of dimensional parameters and \mathbf{P} is the space of positional parameters. Target space \mathbf{E} of mapping f is a finite dimension space. The semantic of space \mathbf{E} is closely related to the modeling of the problem, which does not need to be detailed at the level of abstraction of the present work. The dimensional parameter $u \in \mathbf{U}$ gathers lengths and angles that define the respective shapes of all rigid bodies of the assembly. The positional parameter $p \in \mathbf{P}$ gathers relative or absolute positions of the said rigid bodies of the assembly. Mathematical notation is $(u, p) \mapsto f(u, p)$. The closure equation is then $f(u, p) = 0$. The dimension of space \mathbf{E} is the number of scalar equations of the closure equation. The closure equation captures how rigid bodies are articulated together according to perfect mechanical joints. This generic way to deal with rigid assemblies is very efficient to set up high level concepts.

Consider now a particular solution (u_0, p_0) meaning that $f(u_0, p_0) = 0$. This solution is named the nominal solution because it perfectly fits the closure equation. When the assembly is over-constrained, it is well-known that a small perturbation of the nominal dimension u_0 , noted $u_0 + \varepsilon$, makes the problem impossible, meaning that there exists no positioning parameter p such that $f(u_0 + \varepsilon, p) = 0$. In other words, feeding the closure equation with a non-nominal dimension makes the problem impossible. From the physical point of view, non-nominal components cannot be mounted together, unless some clearance is there, which is addressed in next section.

2.2. Relaxed closure equation of non-nominal assembly

A perfect mechanical joint allows a specified relative motion between two parts. For example, the spherical joint allows any rotation and prevents all translations.

Conversely, a non-perfect mechanical joint allows any relative motion between the two connected parts. Nevertheless, some degrees of freedom are expected to be as wide as the corresponding perfect joint, while remaining degrees of freedom are expected to be very small. For example, the imperfect spherical joint features wide rotation capabilities while translation capabilities feature very small magnitude. The typical example of a non-perfect spherical joint is the nunchaku, a traditional martial arts weapon consisting of two sticks connected at one end by a short chain, as illustrated in Figure 1. Any relative rotation of the sticks is possible but relative translations magnitude is limited by the

chain length.



Figure 1: Non-perfect spherical joint in a “nunchaku”

So, a modeling effort must be done to add flexibility in the joints of the assembly. Formally, this is to add degrees of freedom through additional parameters, noted $j \in \mathbf{J}$, which changes the closure mapping into a relaxed closure mapping $G: \mathbf{U} \times \mathbf{J} \times \mathbf{P} \rightarrow \mathbf{E}$, with $(u, j, p) \mapsto G(u, j, p)$. The relaxed closure mapping is related to the closure mapping through $G(u, 0, p) = f(u, p)$ for all (u, p) . This means that when additional degrees of freedom are inactive, the system is over constrained. The relaxed closure equation is $G(u, j, p) = 0$. In particular, $(u_0, 0, p_0)$ is a solution of the relaxed closure equation.

The point is now that the relaxed closure equation $G(u, j, p) = 0$ is highly under constrained, meaning that given a dimensional parameter u , possibly non nominal, there exists an infinity of parameters j and p such that $G(u, j, p) = 0$.

2.3. Deviation to nominal assembly

By nature, the nominal assembly is a ideal representation that perfectly fulfils the function of the assembly, whatever it can be. So, the deviation to nominal assembly is a reasonable way to quantify how the non-nominal assembly imperfectly fulfils the function.

The first step toward the definition of this deviation is to equip each rigid body of the assembly with characteristic points. For each rigid body, these points are chosen in such a way that a dimensional change results in a change in their relative positions. Furthermore, a dimensional change of one component should also alter the position of neighboring components. Experience shows that EGRMs of joints are good candidates. Formally, let n be the number of rigid bodies.

Body number i is equipped with m_i characteristic points. So, the whole assembly is equipped with the list of characteristic points of its components, noted X_k with $k = 1, \dots, m$ and $m = m_1 + \dots + m_n$. With this notation, points of body number i are X_k with $k = m_1 + \dots + m_{i-1} + 1, \dots, m_1 + \dots + m_{i-1} + m_i$.

Of course, absolute positions of these points depend on the configuration (u, p) of the assembly. This is captured by the notation $X_k(u, p)$ for $k = 1, \dots, m$. So, points of the nominal assembly are $X_k(u_0, p_0)$ for $k = 1, \dots, m$.

The deviation between the nominal assembly and the non-nominal assembly can now be written. Given a dimensional parameter value u , possibly non nominal, and a positional parameter value p , the deviation expression is:

$$D(u, p) = \min_{T \in SE(3)} \frac{1}{2} \sum_{k=1}^m \|X_k(u_0, p_0) - T(X_k(u, p))\|^2$$

To compute the value, the first step is to get the characteristic points of the non-nominal assembly, which are noted $X_k(u, p)$ for $k = 1, \dots, m$. The second step is to compute the best fitting of the non-nominal characteristic points to the nominal characteristic points. Formally, this is to find the rigid motion $T \in SE(3)$ such that previous expression is respected. In the current context, a closed-form solution to this problem is known and explained in [13].

3. Key characteristics

3.1. Minimal clearance

Clearance, a global quantity named J , is a characteristic of the manufactured assembly. It is defined as follows:

$$J(u, p) = \frac{1}{2} \|j\|^2$$

Reader will recognize a norm expression usually used in an optimization context. Maybe other norms can be used, but we have not performed the experiments to certify it.

A first linkage configuration study is to find the positional parameter p such that the additional positional parameters j are as small as possible. This leads to the minimization problem:

$$\begin{cases} J_{min} = \min \{J(u, p)\} \\ G(u, j, p) = 0 \end{cases}$$

Strictly speaking, the minimal clearance J_{min} is not a key characteristic but rather a design constraint, because if $J(u, p)$ is smaller than J_{min} , linkage cannot be assembled and the manufactured system doesn't work. Doing so leads to an assembly featuring joints that are adjusted as much as possible. Could this be a quality criterion? The answer is no. Indeed, among the number of manufactured components of the assembly there exists a possibility that non nominal parts fit together in such a way that joints are almost perfect, despite each component is far from its nominal dimension. So, something must be done to define and quantify the “distance” from the non-nominal assembly to the nominal assembly. The previously defined deviation is chosen to represent this distance. It is the second characteristic of the manufactured assembly.

3.2. Minimal deviation

The minimum deviation between the nominal assembly and the non-nominal one is defined with respect to design or industrial constraints. Constraints are that all local joint clearances $j \in \mathbf{J}$ must be smaller than a constant j^* . For practical reasons, we consider that j^* values are identical for all j . This quantity represents the greatest clearance value accepted by designers or imposed by the manufacturing process. In other words, this is the length of the nunchaku chain previously presented.

Minimal deviation can now be written. Given a dimensional parameter value u , possibly non nominal, the minimal deviation computation is to solve the following constrained optimization problem. Minimal deviation is named D_{min} .

$$\begin{cases} D_{min} = \min \{ D(u, p) \} \\ G(u, j, p) = 0 \\ \{ \forall j \in J, j < j^* \} \end{cases}$$

The computed value D_{min} depends on the dimensional parameter value u and on the maximum value allowed in each joint clearance j^* .

3.3. Maximal deviation

The maximum distance between the nominal assembly and the non-nominal one is defined with respect to design or industrial constraints, in a similar manner than above. Given a dimensional parameter value u , possibly non-nominal, the maximal deviation computation is to solve the following constrained optimization problem. Maximal deviation is named D_{max} .

$$\begin{cases} D_{max} = \max \{ D(u, p) \} \\ G(u, j, p) = 0 \\ \{ \forall j \in J, j < j^* \} \end{cases}$$

Like D_{min} , D_{max} depends on the dimensional parameter value u and on the maximum value allowed in each joint clearance j^* .

3.4. Floating

The fourth and last presented characteristic is floating. This quantity represents the geometrical defect of the mechanical assembly. It quantifies the possible floating phenomenon of the parts of the mechanism. The higher the value is, the more the manufactured parts can move away from their nominal positions, which may cause vibrations and be harmful for the product lifetime.

Given a dimensional parameter value u , possibly non-nominal, floating computation is to subtract D_{min} from D_{max} . Floating is noted F . So:

$$F = D_{max} - D_{min}$$

Obviously, F depends also on the dimensional parameter value u and on the maximum value allowed in each joint clearance j^* .

4. Diagrams

Therefore, we have all the bricks to draw diagrams.

4.1. Parameters u are fixed

The J-D diagram, shown in Figure 2, is sketched for an instance of manufactured mechanical assembly, that is to say where dimensional parameters u are fixed.

This figure illustrates the dependence between the clearance $J(u, p)$ and the deviation $D(u, p)$ as defined above. Four areas are reported.

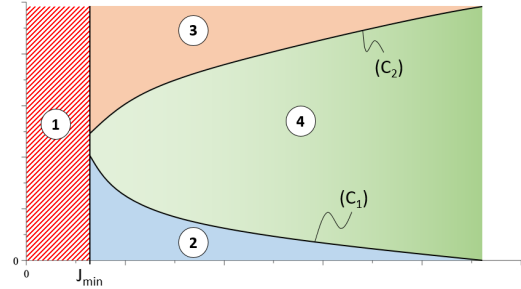


Figure 2: J-D diagram when u is fixed

- Zone 1: The system cannot be assembled because the actual clearance value $J(u, p)$ is less than the J_{min} value.
- Zone 2: The system can be assembled. The deviation value $D(u, p)$ cannot be less than the D_{min} value represented by the (C_1) curve.
- Zone 3: The system can be assembled. The deviation value $D(u, p)$ cannot be greater than the D_{max} value represented by the (C_2) curve.
- Zone 4: This is the area of existence of the system under study. The area represents the allowable floating of the manufactured parts connected together by imperfect joints.

As the reader can appreciate, the D_{max} quantity increases when the clearance $J(u, p)$ increases, while the D_{min} quantity decreases. This behavior is consistent, since the parts can move more freely when imperfect joints allow greater ranges of motion.

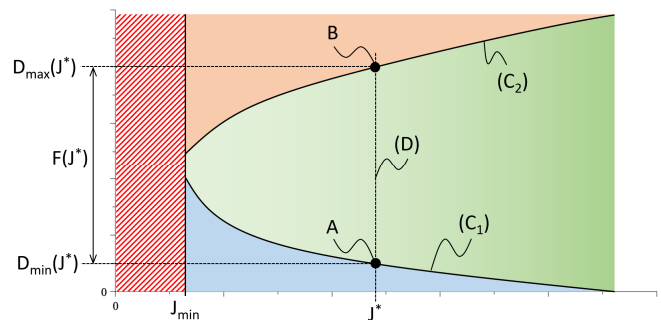


Figure 3: $F(J^*)$ is the floating value for j^* when u is fixed

Figure 3 highlights the construction to obtain the floating value. Given a maximum value j^* for clearance, draw the vertical line (D). Line D and curve (C_1) intersect at point A and line D intersects curve (C_2) at point B. Distance between A and B is the floating value for j^* .

The Figure 4 shows the floating diagram. The (C_3) curve sketched illustrates the floating evolution of the mechanical assembly versus clearance in the imperfect joints. The zone

identified by number 5 is an impossible area for the mechanical assembly under study.

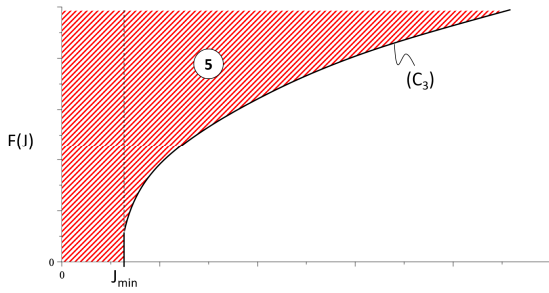


Figure 4: Floating diagram when u is fixed

4.2. Parameters u vary between two limits

The diagram shown in Figure 5 is sketched for a series of assemblies. This means that dimensions of parts vary between two limits imposed by the manufacturing process. The allowed tolerance of the parts are noted u^* . Therefore the dimensional parameters u is such that $|u - u_0| < u^*$. For practical reasons, we consider that u^* values are the same for all u .

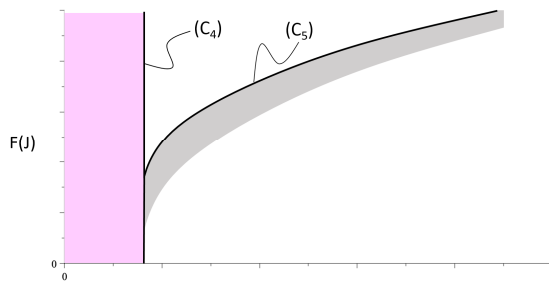


Figure 5: Areas swept by curves C_3 and J_{min} , when $|u - u_0| < u^*$

It is noticeable that for each value of u , the (C_3) curve slightly changes, and that the vertical line J_{min} is shifted a little. The union of all curves defines the gray and pink areas in Figure 5. Curve (C_4) is the right limit of this pink area. Curve (C_5) is the upper limit of the gray area. Analysis of this figure indicates that in order to ensure assemblability and interchangeability of all manufactured parts, the smallest value of J_{min} must be outside of the pink area.

Thanks to this diagram, designers team knows that the floating value of the whole set of manufactured products, will be always under (C_5) curve.

5. Case study

In this section, we applied the presented approach to one case study: a planar mechanism made of six parts and eight joints.

5.1. Presentation

This 2D-assembly is includes six bars linked together by six spherical joints as described in Figure 6.

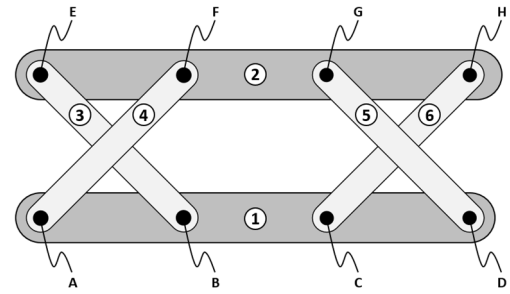


Figure 6: 6-bars structure studied

The nominal length of each horizontal bar is 300 mm. Points **A**, **B**, **C** and **D** (resp. **E**, **F**, **G** and **H**) are aligned and equidistant in bar (1) (resp. bar (2)). The nominal length of each crossbar (numbered from 3 to 6) is 141,4 mm. This mechanical assembly is over-constrained.

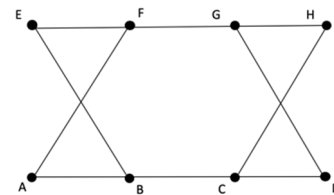


Figure 7: 6-bars structure studied

Classically, conceptual design involves geometry through a skeleton made of points and line segments. Thicknesses of parts are not taken into account. As illustrated in Figure 7, connections are characterized by a point and parts by straight lines.

5.2. Floating diagrams $F(J)$ when $|u - u_0| < u^*$

By varying lengths of parts, a random set of 1000 articulated systems is generated. Then, by using a Maple® prototype program, clearance values and the floating values are computed for each system.

Figure 8 illustrates the resulting curves with tolerance $u^* = 0.05$ mm. Figure 9 illustrates the resulting curves with $u^* = 0.1$ mm. Using a standard Macbook Pro computer, each test requires a computing time of 4362s.

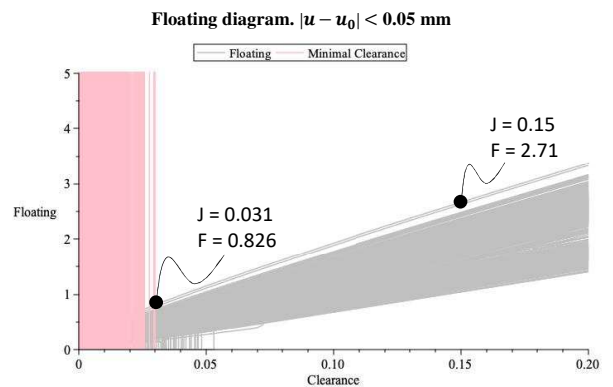


Figure 8: 1st case - Results for $|u - u_0| < 0.05$ mm

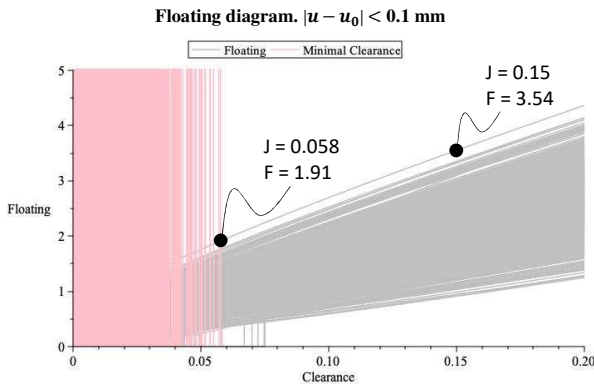


Figure 9: 2nd case - Results for $|u - u_0| < 0.1$ mm

We notice that:

- The maximum value of J_{min} is 0.031mm for an allowed variation of the part dimensions less than 0.05mm.
- The maximum value of J_{min} is 0.058mm for an allowed variation of the part dimensions less than 0.1 mm

Results are consistent. As dimensional uncertainty increases, the largest minimum clearance, insuring assemblability, increases also. Moreover, $Max(J_{min})$ value doubles by doubling u^* . Take a particular value of J , named J^* , for example at $J^* = 0.15$ mm, the floating value is 2.71 for $|u - u_0| < 0.05$ mm while it is 3.54 for $|u - u_0| < 0.1$ mm. When the dimensional uncertainty increases, the floating value also increases but not in the same proportions as the allowed tolerances. The minimum floating value is 0.826 in the first case while it is 1.91 in the second one. Thus, even if we allow tolerances of 0.05 mm, we can never have a floating value lower than 0.826.

Global measures of the mechanical assembly provided by this diagram allow designers to compare different tolerancing solutions. They can predict the floating phenomenon of parts with respect to the manufacturing tolerances of parts.

6. Conclusion

Presented tools allow engineers to study clearances and deviations during the conceptual design phase. Three characteristics of manufactured mechanical assembly are used: clearance, deviation and floating. The last two are new. They are defined and their algebraic expressions are given.

Two diagrams are drawn. The J-D diagram highlights four zones. Three of them are prohibited, the other represents the existence zone of the assembly under study. The floating diagram highlights the dependency between the floating phenomenon and the clearance allowed into the imperfect joints.

All these features provide genuine arguments for selecting the correct level of quality for manufacturing process. They provide also fruitful inputs to compare different architectural solutions of a same product with an equal manufacturing capability level.

The number of input data of the proposed tool is very small: connectivity matrix of the assembly structure,

coordinates of the characteristics points, manufacturing precision class of the parts and range precision of the joint clearance. First simulation results are quickly obtained but the degree of confidence is questionable. Nevertheless, the longer the simulation is, the more reliable it is. For these reasons, our proposal is dedicated to assist designer's team during early phase of development products.

Perspectives of this work are being considered. The first one is to develop a new prototype to experiment other cases. Actually, our platform is implemented with Maple® language for some cases and with the standard optimization command "Minimize". An improvement would be using casADi, an open-source tool for non-linear optimization. A second perspective is a fundamental investigation about how to systematically introduce clearance parameters into perfect joints for standard kinematic 2D and 3D joints. A third perspective is related to deviation local severity. Clearly, not all deviations are equally severe. A concept can be robust to some deviations and sensitive to other deviations, and the present work does not address this point.

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