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A Systematic Approach for Geometrical and Dimensional Tolerancing in Reverse Engineering

George J. Kaisarlis
Rapid Prototyping & Tooling – Reverse Engineering Laboratory
Mechanical Design & Control Systems Section,
School of Mechanical Engineering,
National Technical University of Athens (NTUA)
Greece

1. Introduction

In the fields of mechanical engineering and industrial manufacturing the term Reverse Engineering (RE) refers to the process of creating engineering design data from existing parts and/or assemblies. While conventional engineering transforms engineering concepts and models into real parts, in the *reverse engineering* approach real parts are transformed into engineering models and concepts (Várady et al., 1997). However, apart from its application in mechanical components, RE is a very common practice in a broad range of diverse fields such as software engineering, animation/entertainment industry, microchips, chemicals, electronics, and pharmaceutical products. Despite this diversity, the reasons for using RE appear to be common in all these application fields, e.g. the original design data and documentation of a product are either not updated, not accessible, do not exist or even have never existed. Focusing on the mechanical engineering domain, through the application of RE techniques an existing part is recreated by acquiring its' surface and/ or geometrical features' data using contact or non contact scanning or measuring devices (Wego, 2011). The creation of an RE component computer model takes advantage of the extensive use of CAD/CAM/CAE systems and apparently provides enormous gains in improving the quality and efficiency of RE design, manufacture and analysis. Therefore, RE is now considered one of the technologies that provide substantial business benefits in shortening the product development cycle (Raja & Fernandes, 2008).

Tolerance assignment is fundamental for successful mechanical assembly, conformance with functional requirements and component interchangeability, since manufacturing with perfect geometry is virtually unrealistic. In engineering drawings Geometric Dimensioning and Tolerancing (GD&T) correlates size, form, orientation and location of the geometric elements of the design model with the design intent, therefore it has a profound impact on the manufacturability, ease of assembly, performance and ultimate cost of the component. High geometrical and dimensional accuracy leads to high quality; however, tight tolerances lead to an exponential increase of the manufacturing cost. Though the importance of tolerance design is well understood in the engineering community it still remains an engineering task that largely depends on experimental data, industrial databases and guidelines, past experience and individual expertise, (Kaisarlis et al., 2008). Geometrical and dimensional tolerances are of particular importance, on the other hand, not only in industrial production but also in product development, equipment upgrading and maintenance. The last three activities include, inevitably, RE tasks which go along with the reconstruction of an object CAD model from measured data and have to do with the assignment of dimensional and geometrical manufacturing tolerances to this object. In that context, tolerancing of RE components address a wide range of industrial applications and real-world manufacturing problems such as tolerance allocation in terms of the actual functionality of a prototype assembly, mapping of component experimental design modifications, spare part tolerancing for machines that are out of production or need improvements and no drawings are available, damage repair, engineering maintenance etc.

The objective of remanufacturing a needed mechanical component which has to fit and well perform in an existing assembly and, moreover, has to observe the originally assigned functional characteristics of the product is rather delicate. The objective in such applications is the designation of geometric and dimensional tolerances that match, as closely as possible, to the original *(yet unknown)* dimensional and geometrical accuracy specifications that reveal the original design intend. RE tolerancing becomes even more sophisticated in case that Coordinate Measuring Machines' (CMM) data of a few or just only one of the original components to be reversibly engineered are within reach. Moreover, if operational use has led to considerable wear/ damage or one of the mating parts is missing, then the complexity of the problem increases considerably. The RE tolerancing problem has not been sufficiently and systematically addressed to this date. Currently, in such industrial problems where typically relevant engineering information does not exist, the conventional trial and error approach for the allocation of RE tolerances is applied. This approach apparently requires much effort and time and offers no guarantee for the generation of the best of results.

This research work provides a novel, modern and integrated methodology for tolerancing in RE. The problem is addressed in a systematic, time and cost efficient way, compatible with the current industrial practice. The rest of the chapter is organized as follows: after the review of relevant technical literature in Section 2, the theoretical analysis of RE dimensional and geometrical tolerancing is presented *(Sections 3 and 4 respectively)*. The application of Tolerance Elements (TE) method for cost-effective, competent tolerance designation in RE is then introduced in Section 5. Certain application examples that illustrate the effectiveness of the methodology are further presented and discussed in Section 6. Main conclusions and future work orientation are included in the final Section 7 of the chapter.

2. Literature review

The purpose and the main application areas of RE along with current methodologies and practical solutions for reverse engineering problems in industrial manufacturing are identified and discussed in several reference publications, e.g (Ingle, 1994; Raja & Fernandes, 2008; Wego, 2011). Moreover, the application of RE techniques and their implementation on modern industrial engineering practice is the subject of a numerous research works, e.g. (Abella et al., 1994; Bagci, 2009; Dan & Lancheng, 2006; Endo, 2005; Zhang, 2003). In that context, RE methodologies are applied for the reconstruction of mechanical components and assemblies that have been inevitably modified during several stages of their life cycle, e.g. surface modifications of automotive components during prototype functionality testing (Chant et al., 1998; Yau, 1997), mapping of sheet metal forming deviations on free form surface parts (Yuan et al., 2001), monitoring on the geometrical stability during test runs of mock-up turbine blades used in nuclear power generators (Chen & Lin, 2000), repair time compression by efficient RE modeling of the broken area and subsequent rapid spare part manufacturing (Zheng et al., 2004), recording of die distortions due to thermal effects for the design optimization of fan blades used in aero engines (Mavromihales et al., 2003).

The principles and applications of tolerancing in modern industrial engineering can also be found in several reference publications, e.g. (Drake, 1999; Fischer, 2004). An extensive and systematic review of the conducted research and the state of the art in the field of dimensioning and tolerancing techniques is provided by several recent review papers, e.g. (Singh et al. 2009a, 2009b) and need not be reiterated here. In the large number of research articles on various tolerancing issues in design for manufacturing that have been published over the last years, the designation of geometrical tolerances has been adequately studied under various aspects including tolerance analysis and synthesis, composite positional tolerancing, geometric tolerance propagation, datum establishment, virtual manufacturing, inspection and verification methods for GD&T specifications, e.g. (Anselmetti and Louati, 2005; Diplaris & Sfantsikopoulos, 2006; Martin et al., 2011).

Although RE-tolerancing is a very important and frequently met industrial problem, the need for the development of a systematic approach to extract appropriate design specifications that concern the geometric accuracy of a reconstructed component has been only recently pointed out, (Borja et al., 2001; Thompson et al., 1999; VREPI, 2003). In one of the earliest research works that systematically addresses the RE-tolerancing problem (Kaisarlis et al., 2000), presents the preliminary concept of a knowledge-based system that aims to the allocation of standard tolerances as per ISO-286. The issue of datum identification in RE geometric tolerancing is approached in a systematic way by (Kaisarlis et al, 2004) in a later publication. Recently, (Kaisarlis et al, 2007, 2008) have further extend the research on this area by focusing on the RE assignment of position tolerances in the case of fixed and floating fasteners respectively. The methodology that is presented in this Chapter further develops the approach that is proposed on these last two publications. The novel contribution reported here deals with *(i)* the systematic assignment of *both* geometrical and dimensional tolerances in RE and their possible interrelation through the application of material modifiers on *both* the RE features and datums and *(ii)* the consideration of cost-effective, competent tolerance designation in RE in a systematic way.

3. Dimensional tolerancing in reverse engineering

Reconstructed components must obviously mate with the other components of the mechanical assembly that they belong to, in *(at least)* the same way as their originals, in order the original assembly clearances to be observed, i.e. they must have appropriate manufacturing tolerances. As pointed out in Section 1, this is quite different and much more difficult to be achieved in RE than when designing from scratch where, through normal tolerance analysis/synthesis techniques and given clearances, critical tolerances are assigned, right from the beginning, to all the assembly components. Integration of geometric accuracy constrains aimed at the reconstruction of 3D models of RE-conventional engineering objects from range data has been studied adequately, (Raja & Fernandes, 2008; Várady et al., 1997). These studies deal, however, with the mathematical accuracy of the reconstructed CAD model by fitting curves and surfaces to 3D measured data. Feature–based RE (Thompson et al., 1999; VREPI, 2003) does not address, on the other hand, until now issues related with the manufacturing tolerances which have to be assigned on the CAD drawings in order the particular object to be possible to be made as required. A methodology for the problem treatment is proposed in the following sections.

Engineering objects are here classified according to their shape either as *free-form objects* or as *conventional engineering objects* that typically have simple geometric surfaces *(planes, cylinders, cones, spheres and tori)* which meet in sharp edges or smooth blends. In the following, Feature–Based RE for mechanical assembly components of the latter category is mainly considered. Among features of size (ASME, 2009), *cylindrical features* such as holes in conjunction with pegs, pins or (screw) shafts are the most frequently used for critical functions as are the alignment of mating surfaces or the fastening of mating components in a mechanical assembly. As a result, their role is fundamental in mechanical engineering and, consequently, they should be assigned with appropriate dimensional and geometrical tolerances. In addition, the stochastic nature of the manufacturing deviations makes crucial, for the final RE outcome, the quantity of the available (same) components that serve as reference for the measurements. The more of them are available the more reliable will be the results. For the majority of the RE cases, however, their number is extremely limited and usually ranges from less than ten to only one available item. Mating parts can also be inaccessible for measurements and there is usually an apparent lack of adequate original design and/or manufacturing information. In the scope of this research work, the developed algorithms address the full range of possible scenarios, from "only one original component – no mating component available" to "two or more original pairs of components available", focusing on parts for which either an ISO 286-1 clearance fit (of either hole or shaft basis system) or ISO 2768 (general tolerances) were originally designated.

Assignment of RE dimensional tolerances is accomplished by the present method in five sequential steps. In the primary step (a) the analysis is appropriately directed to ISO fits or general tolerances. In the following steps, the candidate *(Step b)*, suggested *(Step c)* and preferred *(Step d)* sets of RE-tolerances are produced. For the final RE tolerance selection *(Step e)* the cost-effective tolerancing approach, introduced in Section 5, is taken into consideration. For the economy of the chapter, the analysis is only presented for the "*two or more original pairs of components available"* case, focused on ISO 286 fits, as it is considered the most representative.

3.1 Direction of the analysis on ISO fits and/or general tolerances

Let $\boldsymbol{Rd}_{M\_h}$, $\boldsymbol{RF}_h$, $\boldsymbol{RRa}_h$, $\boldsymbol{U}_h$ and $\boldsymbol{Rd}_{M\_s}$, $\boldsymbol{RF}_s$, $\boldsymbol{RRa}_s$, $\boldsymbol{U}_s$ be the sets of the measured diameters, form deviations, surface roughness, and the uncertainty of CMM measurements for the RE-*hole* and the RE-*shaft* features respectively. The Δ_{max}, Δ_{h_max}, Δ_{s_max} limits are calculated by,

$$\Delta_{\max}= \max\{(\max\boldsymbol{Rd}_{M\_h} - \min\boldsymbol{Rd}_{M\_h}), (\max\boldsymbol{Rd}_{M\_s} - \min\boldsymbol{Rd}_{M\_s}), \max\boldsymbol{RF}_h, \max\boldsymbol{RF}_s, \\ (60\cdot\mathrm{mean}\boldsymbol{RRa}_h), (60\cdot\mathrm{mean}\boldsymbol{RRa}_s), \boldsymbol{U}_h, \boldsymbol{U}_s\},$$

$$\Delta_{h\_max} = \max\{(\max \boldsymbol{Rd}_{M\_h} - \min \boldsymbol{Rd}_{M\_h}),\ \max \boldsymbol{RF}_h,\ (60\cdot \mathrm{mean}\boldsymbol{RRa}_h),\ \boldsymbol{U}_h\},$$

$$\Delta_{s\_max} = \max\{(\max \boldsymbol{Rd}_{M\_s} - \min \boldsymbol{Rd}_{M\_s}),\ \max \boldsymbol{RF}_s,\ (60\cdot \mathrm{mean}\boldsymbol{RRa}_s),\ \boldsymbol{U}_s\}$$

In this step, the analysis is directed on the assignment of either ISO 286-1 clearance fits or ISO 2768 general tolerances through the validation of the condition,

$$\Delta_{h\_max} + \Delta_{s\_max} + |a| \geq \max \boldsymbol{Rd}_{M\_h} - \min \boldsymbol{Rd}_{M\_s} \quad (1)$$

where $|a|$ is the absolute value of the maximum ISO 286 clearance Fundamental Deviation (FD) for the relevant nominal sizes range *(the latter is approximated by the mean value of* $\boldsymbol{Rd}_{M\_h}$, $\boldsymbol{Rd}_{M\_s}$ *sets)*. If the above condition is *not* satisfied the analysis is exclusively directed on ISO 2768 general tolerances. Otherwise, the following two cases are distinguished, *(i)* $\Delta_{max} \leq$ IT 11 and *(ii)* IT 11< $\Delta_{max} \leq$ IT 18. In the first case the analysis aims only on ISO 286 fits, whereas in the second case, *both* ISO 286 and ISO 2768 RE tolerances are pursued.

3.2 Sets of candidate IT grades, fundamental deviations and nominal sizes

The starting point for the *Step (b)* of the analysis is the production of the Candidate tolerance grades sets, $\boldsymbol{IT}_{CAN\_h}$, $\boldsymbol{IT}_{CAN\_s}$, for the hole and shaft features respectively. It is achieved by filtering the *initial Candidate IT grades* set, $\boldsymbol{IT}_{CAN\_INIT}$, which includes all standardized IT grades from IT01 to IT18, by the following conditions *(applied for both the h and s indexes),*

$$\begin{array}{ll} IT_{CAN} \geq \max \boldsymbol{RF}, & IT_{CAN} \geq \max \boldsymbol{Rd}_M - \min \boldsymbol{Rd}_M \\ IT_{CAN} \leq 60\cdot \mathrm{mean}\boldsymbol{RRa}, & IT_{CAN} \geq \boldsymbol{U} \end{array} \quad (2)$$

Moreover, in case when estimated maximum and minimum functional clearance limits are available (maxCL, minCL), candidate IT grades are qualified by the validation of,

$$\begin{array}{c} IT_{CAN} < \mathrm{maxCL} \\ IT_{CAN} < \mathrm{maxCL} - \mathrm{minCL} \end{array} \quad (3)$$

The above constraints are applied separately for the hole and shaft and qualify the members of the $\boldsymbol{IT}_{CAN\_h}$, $\boldsymbol{IT}_{CAN\_s}$ sets. Likewise, the set of *initial* Candidate Fundamental Deviations, $\boldsymbol{FD}_{CAN\_INIT}$, that contains all the FDs applicable to clearance fits i.e. $\boldsymbol{FD}_{CAN\_INIT}$ = {a, b, c, cd, d, e, f, fg, g, h}, is filtered by the constraints,

$$FD_{CAN} \leq \min \boldsymbol{Rd}_{M\_h} - \max \boldsymbol{Rd}_{M\_s} \quad (4)$$

$$\begin{array}{c} FD_{CAN} \geq \mathrm{minCL} \\ FD_{CAN} < \mathrm{maxCL} - (\min \boldsymbol{IT}_{CAN\_h} + \min \boldsymbol{IT}_{CAN\_s}) \end{array} \quad (5)$$

The latter constraints, (5), apparently only apply in case of maxCL and/or minCL availability. All qualified FDs are included in the common set of Candidate Fundamental Deviations, $\boldsymbol{FD}_{CAN}$. In the final stage of this step, the Candidate Nominal Sizes Sets, $\boldsymbol{NS}_{CAN\_h}$, $\boldsymbol{NS}_{CAN\_s}$, are initially formulated for the *hole* and *shaft* respectively. Their first members are obtained from the integral part of the following equations,

$$\begin{array}{l} NS_{CAN\_h\_1} = \mathrm{int}\,[\min \boldsymbol{Rd}_{M\_h} - \max FD_{CAN} - \max IT_{CAN\_h}] \\ NS_{CAN\_s\_1} = \mathrm{int}\,[\max \boldsymbol{Rd}_{M\_s} + \max FD_{CAN} + \max IT_{CAN\_s}] \end{array} \quad (6)$$

Following members of the sets are then calculated by an incremental increase, δ, of $NS_{CAN\_h\_1}$ and $NS_{CAN\_s\_1}$,

$$\begin{array}{ll|l} NS_{CAN\_h\_2} = NS_{CAN\_h\_1} + \delta & & NS_{CAN\_s\_2} = NS_{CAN\_s\_1} - \delta \\ NS_{CAN\_h\_3} = NS_{CAN\_h\_2} + \delta & & NS_{CAN\_s\_3} = NS_{CAN\_s\_2} - \delta \\ \cdots\cdots\cdots\cdots & & \cdots\cdots\cdots\cdots \\ NS_{CAN\_h\_v} = NS_{CAN\_h\_v\text{-}1} + \delta & & NS_{CAN\_s\_\mu} = NS_{CAN\_s\_\mu\text{-}1} - \delta \end{array} \tag{7}$$

bounded by,

$$\begin{array}{l} NS_{CAN\_h\_v} \leq \min \boldsymbol{Rd}_{M\_h} \\ NS_{CAN\_s\_\mu} \geq \max \boldsymbol{Rd}_{M\_s} \end{array} \tag{8}$$

with the populations v, μ not necessarily equal. In the relevant application example of section 6, δ is taken $\delta=0.05mm$. Other δ-values can be, obviously, used depending on the case. Since both hole and shaft have a common nominal size in ISO-286 fits, the Candidate Nominal Sizes Set, $\boldsymbol{NS}_{CAN}$, is then produced by the common members of $\boldsymbol{NS}_{CAN\_h}$, $\boldsymbol{NS}_{CAN\_s}$,

$$\boldsymbol{NS}_{\mathrm{CAN}} = \boldsymbol{NS}_{\mathrm{CAN\_h}} \cap \boldsymbol{NS}_{\mathrm{CAN\_s}} \tag{9}$$

3.3 Sets of suggested fits

In Step (c) of the analysis, a combined qualification for the members of the $\boldsymbol{IT}_{\mathrm{CAN\_h}}$, $\boldsymbol{IT}_{\mathrm{CAN\_s}}$, $\boldsymbol{FD}_{CAN}$ and $\boldsymbol{NS}_{\mathrm{CAN}}$ sets is performed in order to produce the two sets of *suggested* Basic Hole, $\boldsymbol{BH}_{\mathrm{SG}}$, and Basic Shaft $\boldsymbol{BS}_{\mathrm{SG}}$ fits. The members of $\boldsymbol{IT}_{\mathrm{CAN\_h}}$ and $\boldsymbol{IT}_{\mathrm{CAN\_s}}$ sets are sorted in ascending order. For the production of the $\boldsymbol{BH}_{\mathrm{SG}}$ set, every candidate nominal size of the $\boldsymbol{NS}_{\mathrm{CAN}}$ set is initially validated against all members of the $\boldsymbol{IT}_{\mathrm{CAN\_h}}$ set, Figure 1(a),

$$\mathrm{NS}_{\mathrm{CAN\_n}} + \mathrm{IT}_{\mathrm{CAN\_h\_\kappa}} \geq \max \boldsymbol{Rd}_{M\_h} \quad \forall \ \mathrm{NS}_{\mathrm{CAN\_n}} \in \boldsymbol{NS}_{\mathrm{CAN}} \tag{10}$$

$\kappa=1, 2, \ldots, i \quad 1\leq i \leq 20$

In case no member of the $\boldsymbol{IT}_{\mathrm{CAN\_h}}$ set satisfies the condition (10) for a particular $\mathrm{NS}_{\mathrm{CAN\_n}}$, the latter is excluded from the $\boldsymbol{BH}_{\mathrm{SG}}$ set. In order to qualify for the $\boldsymbol{BH}_{\mathrm{SG}}$ set, candidate nominal sizes that validate the condition (10) are further confirmed against all members of the $\boldsymbol{FD}_{CAN}$ set, the candidate IT grades of the $\boldsymbol{IT}_{\mathrm{CAN\_s}}$ set and, as well as, the measured RE-shaft data, through the constraints, Figure 1(b),

$$\mathrm{NS}_{\mathrm{CAN\_n}} - \mathrm{FD}_{\mathrm{CAN\_q}} \geq \max \boldsymbol{Rd}_{M\_s} \quad \forall \ \mathrm{FD}_{\mathrm{CAN\_q}} \in \boldsymbol{FD}_{\mathrm{CAN}} \tag{11}$$

$\zeta=1, 2, \ldots, j \qquad 1 \leq j \leq 20$

$$\min \boldsymbol{Rd}_{M\_s} \geq \mathrm{NS}_{\mathrm{CAN\_n}} - \mathrm{FD}_{\mathrm{CAN\_q}} - \mathrm{IT}_{\mathrm{CAN\_s\_\zeta}} \ \forall \ \mathrm{FD}_{\mathrm{CAN\_q}} \in \boldsymbol{FD}_{\mathrm{CAN}} \tag{12}$$

In case no member of the $\boldsymbol{FD}_{CAN}$ set satisfies the condition (11) for a particular $\mathrm{NS}_{\mathrm{CAN\_n}}$, the latter is excluded from the $\boldsymbol{BH}_{\mathrm{SG}}$ set. Moreover, in case no member of the $\boldsymbol{IT}_{\mathrm{CAN\_s}}$ set satisfies the condition (12) for a particular pair of $\mathrm{FD}_{\mathrm{CAN\_q}}$ and $\mathrm{NS}_{\mathrm{CAN\_n}}$, validated by (11), they are both excluded from the $\boldsymbol{BH}_{\mathrm{SG}}$ set. In a similar manner, the production of the suggested Basic Shaft fits set is achieved by the following set of conditions,

$$\min \boldsymbol{Rd}_{M\_s} \geq \mathrm{NS}_{\mathrm{CAN\_n}} - \mathrm{IT}_{\mathrm{CAN\_s\_\zeta}} \ \forall \ \mathrm{NS}_{\mathrm{CAN\_n}} \in \boldsymbol{NS}_{\mathrm{CAN}} \tag{13}$$

$\zeta=1, 2, \ldots, j \quad 1 \leq j \leq 20$

$$\min \boldsymbol{Rd}_{M\_h} \geq NS_{CAN\_n} + FD_{CAN\_q} \ \forall \ FD_{CAN\_q} \in \boldsymbol{FD}_{CAN} \tag{14}$$

$$NS_{CAN\_n} + FD_{CAN\_q} + IT_{CAN\_h\_\kappa} \geq \max \boldsymbol{Rd}_{M\_h} \ \forall \ FD_{CAN\_q} \in \boldsymbol{FD}_{CAN} \tag{15}$$

$\kappa=1, 2, \ldots, i \quad 1\leq i \leq 20$

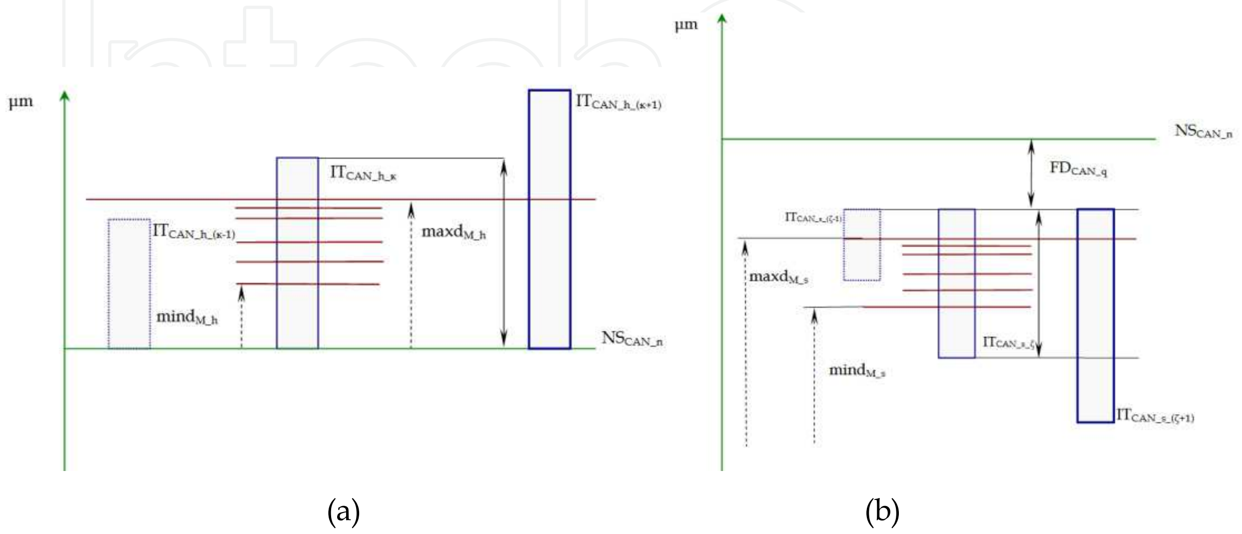

Fig. 1. Suggested Basic Hole fits qualification

3.4 Sets of preferred fits

A limited number of *Preferred Fits* out of the *Suggested* ones is proposed in Step (d) through the consideration of ISO proposed fits. Moreover, the implementation of manufacturing guidelines, such as the fact that it is useful to allocate a slightly larger tolerance to the hole than the shaft, preference of Basic Hole fits over Basic Shaft ones, preference of nominal sizes that are expressed in integers or with minimum possible decimal places etc, are additionally used to "filter" the final range of the preferred fits. The final selection, Step (e), out of the limited set of preferred fits and the method end result is reached by the consideration of the machine shop capabilities and expertise in conjunction with the application of the cost – effective RE tolerancing approach, presented in Section 5 of the chapter.

4. Geometrical tolerancing in reverse engineering

In order to observe interchangeability, *geometrical* as well as dimensional accuracy specifications of an RE component must comply with those of the mating part(-s). GD&T in RE must ensure that a reconstructed component will fit and perform well without affecting the function of the specific assembly. The methodology that is presented in this section focuses on the RE assignment of the main type of geometrical tolerance that is used in industry, due to its versatility and economic advantages, the True Position tolerance. However, the approach can be easily adapted for RE assignment of other location geometrical tolerances types, such as coaxiality or symmetry and, as well as, for run-out or profile tolerances.

Position tolerancing is standardized in current GD&T international and national standards, such as (ISO, 1998; ISO 1101, 2004; ASME, 2009). Although the ISO and the ASME tolerancing systems are not fully compatible, they both define position geometrical tolerance as the total permissible variation in the location of a feature about its exact true position. For cylindrical features such as holes or bosses the position tolerance zone is usually the diameter of the cylinder within which the axis of the feature must lie, the center of the tolerance zone being at the exact true position, Figure 2, whereas for size features such as slots or tabs, it is the total width of the tolerance zone within which the center plane of the feature must lie, the center plane of the zone being at the exact true position. The position tolerance of a feature is denoted with the size of the diameter of the cylindrical tolerance zone (or the distance between the parallel planes of the tolerance zone) in conjunction with the theoretically exact dimensions that determine the true position and their relevant datums, Figure 2. Datums are, consequently, fundamental building blocks of a positional tolerance frame in positional tolerancing. Datum features are chosen to position the toleranced feature in relation to a Cartesian system of three mutually perpendicular planes, jointly called Datum Reference Frame (DRF), and restrict its motion in relation to it. Positional tolerances often require a three plane datum system, named as primary, secondary and tertiary datum planes. The required number of datums (1, 2 or 3) is derived by considering the degrees of freedom of the toleranced feature that need to be restricted. Change of the datums and/or their order of precedence in the DRF results to different geometrical accuracies, (Kaisarlis et al., 2008).

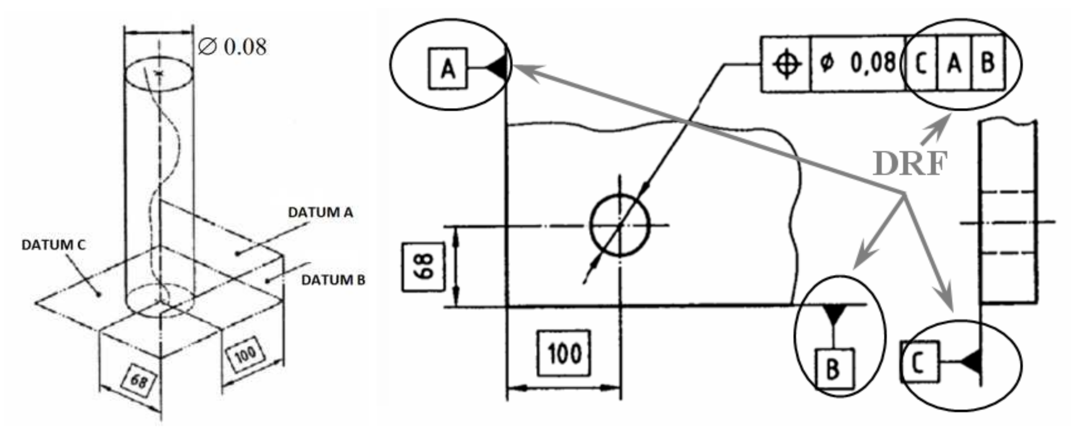

Fig. 2. Cylindrical tolerance zone and geometric true position tolerancing for a cylindrical feature according to ISO 1101 (Kaisarlis et al, 2008)

The versatility and economic benefits of true position tolerances are particularly enhanced when they are assigned at the Maximum Material Condition (MMC). At MMC, an increase in position tolerance is allowed, equal to the departure of the feature from the maximum material condition size, (ISO, 1988; ASME, 2009). As a consequence, a feature with size beyond maximum material but within the dimensional tolerance zone and its axis lying inside the enlarged MMC cylinder is acceptable. The accuracy required by a position tolerance is thus relaxed through the MMC assignment and the reject rate reduced. Moreover, according to the current ISO and ASME standards, datum features of size that are included in the DRF of position tolerances can also apply on either MMC, Regardless of Feature Size (RFS) or Least Material Condition (LMC) basis.

Position tolerances mainly concern clearance fits. They achieve the intended function of a clearance fit by means of the relative positioning and orientation of the axis of the true geometric counterpart of the mating features with reference to one, two or three Cartesian datums. The relationship between mating features in such a clearance fit may be classified either as a fixed or a floating fastener type, (ASME, 2009; Drake, 1999), Figure 3. Floating fastener situation exists where two or more parts are assembled with fasteners such as bolts and nuts, and all parts have clearance holes for the bolts. In a fixed fastener situation one or more of the parts to be assembled have clearance holes and the mating part has restrained fasteners, such as screws in threaded holes or studs. The approach that is here presented deals with *both* the floating and fixed fastener cases by integrating the individual case methodologies published by (Kaisarlis et al. 2007; 2008).

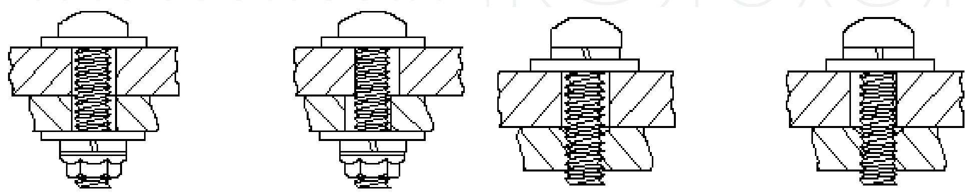

Fig. 3. Typical floating and fixed fasteners and worst case assembly conditions (Drake, 1999)

Basic issues of the assignment of a Position Tolerance in RE are included in Table 1. Limited number of reference components that does not allow for statistical analysis, availability or not of the mating parts and the presence of wear may affect the reliability of the RE results. Moreover, datum selection and the size of the position tolerance itself should ensure, obviously, a stress-free mechanical mating. The analytic approach presented in this section deals with the full range of these issues in order to produce a reliable solution within realistic time.

i.	*The number of available RE components that will be measured. The more RE parts are measured, the more reliable will be the extracted results. Typically, the number of available RE components is extremely limited, usually ranging from less than ten to a single one article.*
ii.	*Off the shelf, worn or damaged RE components. Off the shelf RE components are obviously ideal for the job, given that the extent of wear or damage is for the majority of cases difficult to be quantified or compensated.*
iii.	*Accessibility of the mating part (-s).*
iv.	*Existence of repetitive features in the RE component that may have the same function (group or pattern of features).*
v.	*Type of assembly (e.g. floating or fixed fasteners).*
vi.	*The size and the form (cylindrical, circular, square, other) of the geometrical tolerance zone.*
vii.	*Candidate datums and datum reference frames. Depending on the case more possible DRFs may be considered.*
viii.	*Precedence of datum features in DRFs.*
ix.	*Theoretical (basic) dimensions involved.*
x.	*Assignment of Maximum Material and Least Material Conditions to both the RE-feature and RE datum features.*
xi.	*Measurement instrumentation capabilities in terms of final uncertainty of the measurements results. Measurements methods and software.*

Table 1. Issues of Geometrical Tolerance Assignment in RE

Assignment of RE-position tolerance for both the fixed and the floating fastener case is accomplished by the present method in five sequential steps. The analysis is performed individually for each feature that has to be toleranced in the RE-component. At least two RE reference components, intact or with negligible wear, need to be available in order to minimize the risk of measuring a possibly defective or wrongly referenced RE component and, as it is later explained in this section, to improve the method efficiency. This does not certainly mean that the method cannot be used even when only one component is available. Mating part availability is desirable as it makes easier the datum(s) recognition. Minimum assembly clearance and, as well as, the dimensional tolerance of the RE-feature *(hole, peg, pin or screw shaft)* and RE-Datums *(for features of size)* are taken as results from the RE dimensional tolerance analysis presented in the previous section of the chapter in conjunction with those quoted in relevant application- specific standards.

The primary step (a) of the analysis concerns the recognition of the critical features on the RE component that need to be toleranced and, as well as, their fastening situation. This step is performed interactively and further directs the analysis on either the fixed or the floating fastener option. In step (b) mathematical relationships that represent the geometric constraints of the problem are formulated. They are used for the establishment of an initial *set of candidate position tolerances.* The next step (c) qualifies *suggested sets* out of the group (b) that have to be in conformance with the measured data of the particular RE-feature. The step (d) of the analysis produces then a set of *preferred position tolerances* by filtering out the output of step (c) by means of knowledge-based rules and/or guidelines. The capabilities and expertise of the particular machine shop, where the new components will be produced, and the cost-tolerance relationship, are taken into consideration in the last step (e) of the analysis, where the required position tolerance is finally obtained. For every *datum feature* that can be considered for the position tolerance assignment of an *RE-feature*, the input for the analysis consists of *(i)* the measured form deviation of the datum feature (e.g. flatness), *(ii)* its measured size, in case that the datum is a feature of size (e.g. diameter of a hole) and *(iii)* the orientation deviation (e.g. perpendicularity) of the RE-feature axis of symmetry with respect to that datum. The orientation deviations of the latter with respect to the two other datums of the same DRF have also to be included (perpendicularity, parallelism, angularity). Input data relevant with the RE-feature itself include its measured size (e.g. diameter) and coordinates, e.g. X, Y measured dimensions by a CMM, that locate its axis of symmetry. Uncertainty of the measured data should conform to the pursued accuracy level. In that context the instrumentation used for the measured input data, e.g. ISO 10360-2 accuracy threshold for CMMs, is considered appropriate for the analysis only if its uncertainty is at six times less than the minimum assembly clearance.

4.1 Sets of candidate position tolerance sizes

The size of the total position tolerance zone is determined by the minimum clearance, min*CL*, of the (hole, screw-shaft) assembly. It ensures that mating features will assemble even at worst case scenario, i.e. when both parts are at MMC and located at the extreme ends of the position tolerance zone (ASME, 2009). The equations (16 -i) and (16-ii) apply for the fixed and floating fastener case respectively,

$$\begin{gathered} (i)\ \ T_{POS} = \text{minCL} = T_{POS\_s} + T_{POS\_h} = MMC_h - MMC_s \\ (ii)\ \ T_{POS} = \text{minCL} = T_{POS\_h} \end{gathered} \tag{16}$$

For the fixed fasteners case, in industrial practice the *total position tolerance* T_{POS} of equation (16-i) is distributed between shaft and hole according to the ease of manufacturing, production restrictions and other factors that influence the manufacturing cost of the mating parts. In conformance with that practice a set of 9 candidate sizes for the position tolerance of the RE-shaft, $\boldsymbol{R}_{CAN\_s}$ and/ or the RE-hole, $\boldsymbol{R}_{CAN\_h}$, is created by the method with a (T_{POS} /10**)** step, which includes the 50% -50% case,

$$\begin{gathered} \boldsymbol{R}_{CAN\_h} = \boldsymbol{R}_{CAN\_s} = \{T_{POS1}, T_{POS2}, \ldots, T_{POSi}, \ldots, T_{POS9}\} \\ \text{where,} \quad T_{POSi} = i \cdot T_{POS}/10, \qquad i=1, 2, \ldots, 9 \end{gathered} \tag{17}$$

For the floating fasteners case the *total position tolerance* T_{POS} of equation (16-ii) actually concerns only RE-features of the Hole type. Therefore, the $\boldsymbol{R}_{CAN\_h}$ set only contains the T_{POS} element. The above tolerances attain, apparently, their maximum values when the RE feature own dimensional tolerance zone is added,

$$\begin{gathered} T_{POSi\_MAX} = T_{POSi} + LMC_h - MMC_h \quad \text{(RE-feature / Hole)} \\ T_{POSi\_MAX} = T_{POSi} + MMC_s - LMC_s \quad \text{(RE- feature / Shaft)} \end{gathered} \tag{18}$$

4.2 Sets of candidate DRFs and theoretical dimensions

To ensure proper RE-part interfacing and safeguard repeatability, datum features of the original part and those of the RE-part should, ideally, coincide. In order to observe this principle the original datum features and their order of precedence have to be determined. Initial recognition of datum features among the features of the RE-part is performed interactively following long established design criteria for locating or functional surfaces and the same, and taking into consideration the mating parts function. Out of all candidate recognized datums an initial set of candidate DRFs, $\boldsymbol{D}_{CAN\_INIT}$, is produced by taking all combinations in couples and in triads between them. A valid DRF should conform with the constraints that have to do with the arrangement and the geometrical deviations of its datums. Only DRFs that arrest all degrees of freedom of the particular RE-feature and consequently have three or at least two datums are considered. DRF qualification for geometric feasibility is verified by reference to the list of the valid geometrical relationships between datums as given in (ASME, 1994). The geometric relationship for instance, for the usual case of three datum planes that construct a candidate DRF is in this way validated, i.e. the primary datum not to be parallel to the secondary and the plane used as tertiary datum not to be parallel to the line constructed by the intersection of the primary and secondary datum planes. Planar or axial datum features are only considered by the method as primary when the axis of the RE-feature is perpendicular in the first case or parallel, in the second one, to them.

The following analysis applies for both the hole and the shaft and is common for the fixed and floating fasteners case. Consequently, the indexes "*h*" or "*s*" are not used hereafter. It is here also noted that the index "*i*" only concerns the fixed fastener case. For the floating fastener case the index "*i*" has a constant value of 1. Let $\boldsymbol{RF}_{DF}$ be the set of the measured form deviations of a candidate datum feature and $\boldsymbol{RO}$ the orientation deviations of the RE feature axis of symmetry with respect to that datum. Fitness of the members of the initial DRF set, $\boldsymbol{D}_{CAN\_INIT}$, against the members of the $\boldsymbol{R}_{CAN}$ set of candidate position tolerance sizes is confirmed regarding the primary datum through the following constraints,

$$\max(\boldsymbol{RF}_{DF}) \leq T_{POSi} \tag{19}$$

$$\max(\boldsymbol{RO}) \leq T_{POSi} \tag{20}$$

Mutual orientation deviations of the secondary and/or tertiary datums, $\boldsymbol{RO}_{DF}$, in a valid DRF should also conform with the position tolerance of equation (16),

$$\begin{array}{ll} \max(\boldsymbol{RF}_{DF}) \leq k \cdot T_{POSi}, & \max(\boldsymbol{RO}) \leq k \cdot T_{POSi} \\ \max(\boldsymbol{RO}_{DF}) \leq k \cdot T_{POSi}, & k \geq 1 \end{array} \tag{21}$$

where k is a weight coefficient depending on the accuracy level of the case. A set of *Candidate* DRFs is thus created, $\boldsymbol{D}_{CAN}{}^{(i)}$, that is addressed to each i member (i=1,…9) of the $\boldsymbol{R}_{CAN}$ set.

Sets of *Candidate Theoretical Dimensions*, $[(\boldsymbol{C}_{CAN}{}^{(ij)}X,\ \boldsymbol{C}_{CAN}{}^{(ij)}Y),\ i=1,2,\ldots 9,\ j=1,2,\ldots,n]$, which locate the RE feature axis of symmetry with reference to every one of the n candidate $DRF^{(i)}{}_j$ of the $\boldsymbol{D}_{CAN}{}^{(i)}$ set are generated at the next stage of the analysis. Measured, from all the available RE reference parts, *axis location coordinate*s are at first integrated into sets, $[\boldsymbol{C}_{CAN}{}^{(ij)}X_M,\ \boldsymbol{C}_{CAN}{}^{(ij)}Y_M]$. Sets of Candidate Theoretical Dimensions are then produced in successive steps starting from those calculated from the integral part of the difference between the minimum measured coordinates and the size of the position tolerance, T_{POSi},

$$X^{(ij)}{}_1 = \mathrm{int}[\min(\boldsymbol{C}_{CAN}{}^{(ij)}X_M) - T_{POSi}],\ Y^{(ij)}{}_1 = \mathrm{int}[\min(\boldsymbol{C}_{CAN}{}^{(ij)}Y_M) - T_{POSi}] \tag{22}$$

Following members of the $\boldsymbol{C}_{CAN}{}^{(ij)}X$, $\boldsymbol{C}_{CAN}{}^{(ij)}Y$ sets are calculated by an incremental increase δ of the theoretical dimensions $X^{(ij)}{}_1$, $Y^{(ij)}{}_1$,

$$\begin{array}{ll} X^{(ij)}{}_2 = X^{(ij)}{}_1 + \delta, & Y^{(ij)}{}_2 = Y^{(ij)}{}_1 + \delta \\ X^{(ij)}{}_3 = X^{(ij)}{}_2 + \delta, & Y^{(ij)}{}_3 = Y^{(ij)}{}_2 + \delta \\ \ldots\ldots\ldots\ldots & \ldots\ldots\ldots\ldots \\ X^{(ij)}{}_p = X^{(ij)}{}_{(p-1)} + \delta, & Y^{(ij)}{}_q = Y^{(ij)}{}_{(q-1)} + \delta \end{array} \tag{23}$$

where as upper limit is taken that of the maximum measured $X^{(ij)}{}_M$, $Y^{(ij)}{}_M$ coordinates plus the position tolerance T_{POSi},

$$X^{(ij)}{}_p \leq \max(\boldsymbol{C}_P{}^{(ij)}X_M) + T_{POSi},\quad Y^{(ij)}{}_q \leq \max(\boldsymbol{C}_P{}^{(ij)}Y_M) + T_{POSi} \tag{24}$$

with the populations p, q of the produced $\boldsymbol{C}_{CAN}{}^{(ij)}X$ *and* $\boldsymbol{C}_{CAN}{}^{(ij)}Y$ sets of candidate theoretical dimensions not necessarily equal. In the case study that is presented δ=0.05mm. Other δ-values can be used as well.

4.3 Sets of suggested position tolerances

Sets of *Suggested DFR*s that are produced in step (b), $\boldsymbol{D}_{SG}{}^{(i)}$, are qualified as subgroups of the sets of *Candidate DFR*s, $\boldsymbol{D}_{CAN}{}^{(i)}$, in accordance with their conformance with the measured location coordinates and the application or not of the Maximum or Least Material Conditions to the RE-feature size or to the RE-Datum size. In conjunction with equation (16), qualification criterion for the Suggested DFR's, $DRF^{(i)}{}_j$ j=1,2,…, n, is, Figure 4(a),

$$\max\{\Delta X^{(ij)}{}_M,\ \Delta Y^{(ij)}{}_M\} \leq T_{POSi} \tag{25}$$

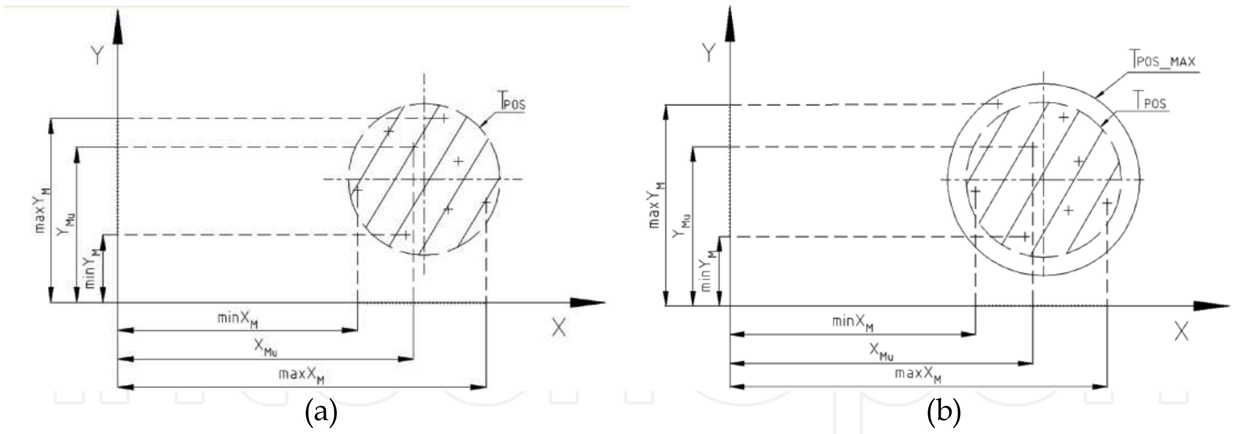

Fig. 4. Qualification conditions for suggested DRFs (Kaisarlis et al., 2007)

where,

$$\begin{aligned} \Delta X^{(ij)}_M &= \max(\boldsymbol{C}_{CAN}{}^{(ij)}X_M) - \min(\boldsymbol{C}_{CAN}{}^{(ij)}X_M) \\ \Delta Y^{(ij)}_M &= \max(\boldsymbol{C}_{CAN}{}^{(ij)}Y_M) - \min(\boldsymbol{C}_{CAN}{}^{(ij)}Y_M) \end{aligned} \tag{26}$$

In case constraint (25) is not satisfied, a $DRF^{(i)}_j$ can only be further considered, when Maximum or Least Material Conditions are applied to RE-feature size, Figure 4(b),

$$\max\{\Delta X^{(ij)}_M, \Delta Y^{(ij)}_M\} \leq T_{POSi\_MAX} \tag{27}$$

In case no member of a $\boldsymbol{D}_{CAN}{}^{(i)}$ (i=1,2,…9) set satisfies either constraint (25) or constraint (27) the relevant T_{POSi} is excluded from the set of Suggested Position Tolerance Sizes, $\boldsymbol{R}_{SG}$.

Let r be the number of the available RE-parts. Sets of *Suggested Theoretical Dimensions*, $[\boldsymbol{C}_{SG}{}^{(ij)}X, \boldsymbol{C}_{SG}{}^{(ij)}Y]$, can now be filtered out of the Candidate Theoretical Dimensions through the application of the relationships,

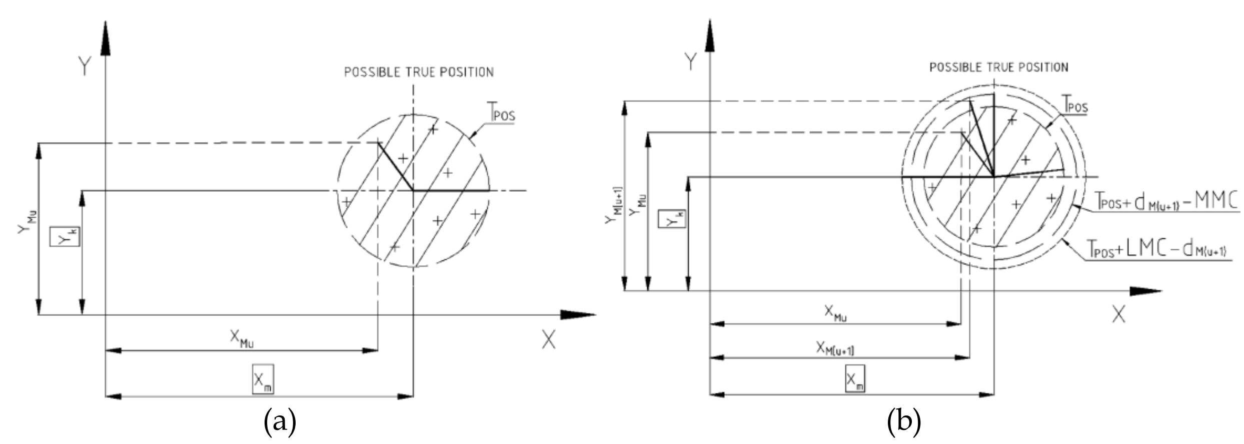

Fig. 5. Qualification conditions for suggested theoretical dimensions (Kaisarlis et al., 2007)

$$|\,X^{(ij)}_m - X^{(ij)}_{Mu}\,| \leq \frac{T_{POSi\_MAX}}{2}, \quad |\,Y^{(ij)}_k - Y^{(ij)}_{Mu}\,| \leq \frac{T_{POSi\_MAX}}{2} \tag{28}$$

m=1,2, …,p ; k=1,2,…,q ; u=1,2,…,r

and the constraint imposed by the geometry of a position tolerance, Figure 5(a),

$$(X^{(ij)}_m - X^{(ij)}_{Mu})^2 + (Y^{(ij)}_k - Y^{(ij)}_{Mu})^2 \le (\frac{T_{POSi}}{2})^2 \quad (29)$$

m=1,2, …,p ; k=1,2,…,q ; u=1,2,…,r

Candidate Theoretical Dimensions that satisfy the constraints (28) but not the constraint (29) can apparently be further considered in conjunction with constraint (27) when Maximum or Least Material Conditions are used. In these cases they are respectively qualified by the conditions, e.g. for the case of RE-feature /Hole, Figure 5(b),

$$(X^{(ij)}_m - X^{(ij)}_{Mu})^2 + (Y^{(ij)}_k - Y^{(ij)}_{Mu})^2 \le (\frac{T_{POSi} + d_{Mu} - MMC}{2})^2 \quad (30)$$

$$(X^{(ij)}_m - X^{(ij)}_{Mu})^2 + (Y^{(ij)}_k - Y^{(ij)}_{Mu})^2 \le (\frac{T_{POSi} + LMC - d_{Mu}}{2})^2 \quad (31)$$

m=1,2, …,p ; k=1,2,…,q ; u=1,2,…,r

When applicable, the case of MMC or LMC on a RE-Datum feature of size may be also investigated. For that purpose, the size tolerance of the datum, T_{S_DF}, must be added on the right part of the relationships (27) and (28). In that context, the constraints (30) and (31), e.g. for the case of RE-feature /Hole - RE-Datum /Hole on MMC, are then formulated as,

$$(X^{(ij)}_m - X^{(ij)}_{Mu})^2 + (Y^{(ij)}_k - Y^{(ij)}_{Mu})^2 \le (\frac{T_{POSi} + d_{Mu} - MMC + d_{Mu\_DF} - MMC_{DF}}{2})^2 \quad (32)$$

$$(X^{(ij)}_m - X^{(ij)}_{Mu})^2 + (Y^{(ij)}_k - Y^{(ij)}_{Mu})^2 \le (\frac{T_{POSi} + LMC - d_{Mu} + d_{Mu\_DF} - MMC_{DF}}{2})^2 \quad (33)$$

m=1,2, …,p ; k=1,2,…,q ; u=1,2,…,r

where d_{Mu_DF} is the measured diameter of the datum on the *u-th* RE-part and MMC_{DF} the MMC size of the RE-Datum.

4.4 Sets of preferred position tolerances

The next step of the analysis provides for three tolerance selection options and the implementation of manufacturing guidelines for datum designation in order the method to propose a limited number of *Preferred Position Tolerance Sets* out of the *Suggested* ones and hence lead the final decision to a rational end result. The first tolerance selection option is only applicable in the fixed fasteners case and focuses for a maximum tolerance size of a $T_{POS}/2$. The total position tolerance T_{POS}, whose distribution between the mating parts is unknown, will be unlikely to be exceeded in this way and therefore, even in the most unfavourable assembly conditions interference will not occur. The second selection option gives its preference to Position Tolerance Sets that are qualified regardless of the application of the Maximum or Least Material Conditions to the RE-feature size and/ or the RE- datum feature size i.e. through their conformance only with the constraint (29) and not the constraints (30) to (33). Moreover, guidelines for datums which are used in the above context are, (ASME 2009; Fischer, 2004):

- A datum feature should be: *(i)* visible and easily accessible, *(ii)* large enough to permit location/ processing operations and *(iii)* geometrically accurate and offer repeatability to prevent tolerances from stacking up excessively

- Physical surfaces are preferable datum features over derived and/ or virtual ones.
- External datums are preferred over internal ones.
- For the fixed fastener case, a preferred primary datum of the mating parts is their respective planar contact surface.
- Theoretical dimensions and tolerances expressed in integers or with minimum possible decimal places are preferable.

5. Cost - effective RE-tolerance assignment

To assign cost optimal tolerances to the new RE-components, that have to be re-manufactured, the Tolerance Element (TE) method is introduced. Accuracy cost constitutes a vital issue in production, as tight tolerances always impose additional effort and therefore higher manufacturing costs. Within the frame of further development of the CAD tools, emphasis is recently given on techniques that assign mechanical tolerances in terms not only of quality and functionality but also of minimum manufacturing cost. Cost-tolerance functions, however, are difficult to be adopted in the tolerance optimization process because their coefficients and exponents are case-driven, experimentally obtained, and they may well not be representative of the manufacturing environment where the production will take place. The TE method (Dimitrellou et al., 2007a; 2007c, 2008) overcomes the mentioned inefficiencies as it automatically creates and makes use of appropriate cost-tolerance functions for the assembly chain members under consideration. The latter is accomplished through the introduction of the concept of Tolerance Elements (Dimitrellou et al., 2007b) that are geometric entities with attributes associated with the accuracy cost of the specific machining environment where the components will be manufactured.

The accuracy cost of a part dimension depends on the process and resources required for the production of this dimension within its tolerance limits. Given the workpiece material and the tolerances, the part geometrical characteristics such as shape, size, internal surfaces, feature details and/or position are taken into consideration for planning the machining operations, programming the machine tools, providing for fixtures, etc. These characteristics have thus a direct impact on the machining cost of the required accuracy and determine, indirectly, its magnitude. A Tolerance Element (TE) is defined either as a 3D form feature of particular shape, size and tolerance, or as a 3D form feature of particular position and tolerance (Dimitrellou et al., 2007a). It incorporates attributes associated with the feature shape, size, position, details and the size ratio of the principal dimensions of the part to which it belongs. For a given manufacturing environment (machine tools, inspection equipment, supporting facilities, available expertise) to each TE belongs one directly related with this environment cost-tolerance function.

TEs are classified through a five-class hierarch system, Figure 6. Class level attributes are all machining process related, generic and straightforwardly identifiable in conformance with the existing industrial understanding. In first level, TEs are classified according to the basic geometry of the part to which they belong, i.e. rotational TEs and prismatic TEs. Rotational TEs belong to rotational parts manufactured mainly by turning and boring, while prismatic TEs belong to prismatic parts mainly manufactured by milling. The contribution of the geometrical configuration of the part to the accuracy cost of a TE, is taken into account in the second level through the size ratio of the principal dimensions of the part. In this way TEs are classified as short [L/D ≤3] and long [L/D >3] TEs, following a typical way of classification. In third level TEs are classified to external and internal ones as the achievement of internal tolerances usually results to higher accuracy costs. The fourth TE classification level distinguishes between plain and complex TEs depending on the absence or presence of additional feature details (grooves, wedges, ribs, threads etc). They do not change the principal TE geometry but they indirectly contribute to the increase of the accuracy cost. In the final fifth level, the involvement of the TE size to the accuracy cost is considered. TEs are classified, to the nominal size of the chain dimension, into six groups by integrating two sequential ISO 286-1 size ranges.

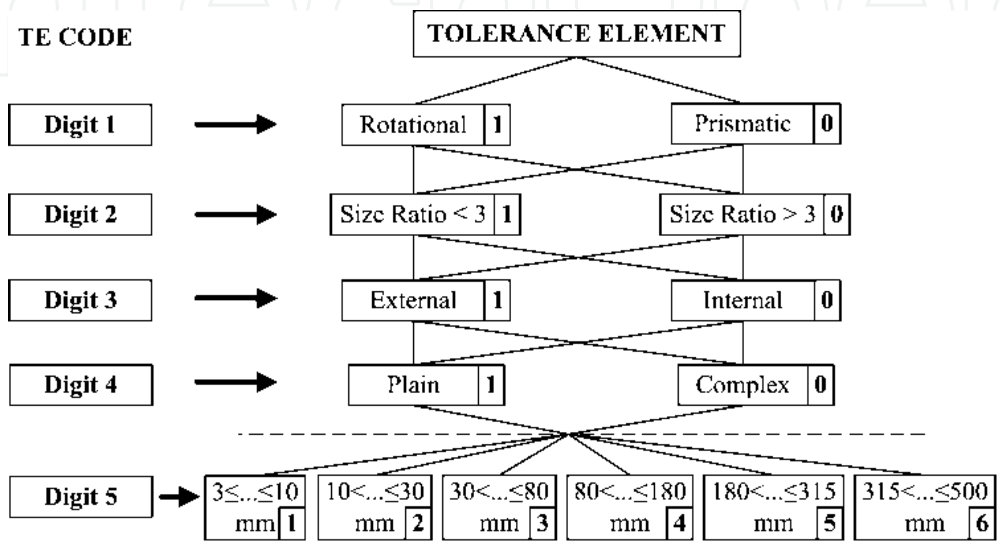

Fig. 6. Tolerance Elements five-class hierarch system (Dimitrellou et al., 2007a)

Based on the TE-method the actual machining accuracy capabilities and the relative cost per TE-class of a particular machine shop are recorded through an appropriately developed Database Feedback Form (DFF). The latter includes the accuracy cost for all the 96 TE-classes in the size range 3-500 mm and tolerances range IT6-IT10. DFF is filled once, at the system commissioning stage, by the expert engineers of the machine shop where the assembly components will be manufactured and can then be updated each time changes occur in the shop machines, facilities and/or expertise. The DFF data is then processed by the system through the least-square approximation and the system constructs and stores a cost-tolerance relationship of the power function type, per TE-class.

5.1 Tolerance chain constrains

In a n-member dimensional chain the tolerances of the individual dimensions $D_1, D_2, \ldots, D_n$, control the variation of a critical end-dimension D_0, according to the chain,

$$D_0 = f(D_1, D_2, \ldots, D_n) \tag{34}$$

where f(D) can be either a linear or nonlinear function. To ensure that the end-dimension will be kept within its specified tolerance zone, the worst-case constrain that provides for 100% interchangeability has to be satisfied,

$$f\left(D_{i\max} + D_{j\min}\right) \le D_0 + t_0 \tag{35}$$

$$D_0 - t_0 \leq f\left(D_{i\min} + D_{j\max}\right) \tag{36}$$

where t_0 is the tolerance of the end-dimension D_0. For an (i+j)-member dimensional chain dimensions D_i constitute the positive members of the chain while dimensions D_j constitute its negative members. In RE tolerancing, preferred alternatives for nominal sizes and dimensional tolerances that are generated from the analysis of Section 3, for each dimension involved in the chain are further filtered out by taking into consideration the above tolerance chain constraints.

5.2 Minimum machining cost

A second sorting out is applied by taking into account the accuracy cost for each combination of alternatives that obtained in the previous stage. Cost-tolerance functions are provided by the machine shop DFF and the total accuracy cost is thus formulated as,

$$C_{total} = \sum_{i=1}^{n} C\left(t_i\right) = \sum_{i=1}^{n} \left[A_i + B_i \,/\, t_i^{k_i} \right] \rightarrow \min \tag{37}$$

where C(t) is the relative cost for the production of the machining tolerance ±t and A, B, k are constants. The combination of alternatives that corresponds to the minimum cost is finally selected as the optimum one.

6. Application examples and case studies

In order to illustrate the effectiveness of the proposed method three individual industrial case studies are presented in this section. All necessary input data measurements were performed by means of a direct computer controlled CMM (*Mistral*, Brown & Sharpe-DEA) with ISO 10360-2 max. permissible error 3.5μm and PC-DMIS measurement software. A Renishaw PH10M head with TP200 probe and a 10mm length tip with diameter of 2mm were used. The number and distribution of sampling points conformed with the recommendations of BS7172:1989, (Flack, 2001), (9 points for planes and 15 for cylinders).

6.1 Application example of RE dimensional tolerancing

For a reverse engineered component of a working assembly (Part 2, Figure 7) assignment of dimensional tolerances was carried out using the developed methodology. The case study assembly of Figure 7 is incorporated in an optical sensor alignment system. Its' location, orientation and dynamic balance is considered of paramount importance for the proper function of the sensor. The critical assembly requirements that are here examined are the clearance gaps between the highlighted features (D1, D2, D3) of Part 1 – Shaft and Part 2-Hole in Figure 8. Four intact pairs of components were available for measurements. The analysis of section 3 was performed for all three critical features of Figure 8 individually. However, for the economy of the chapter, input data and method results are only presented for the D2 RE-feature, in Tables 2 and 3 respectively. The selected ISO 286 fits, Figure 9, produced in 12min *(10min CMM-measurements + 2min Computer aided implementation)* were experimentally verified and well approved by fitting reconstructed components in existing and in use assemblies.

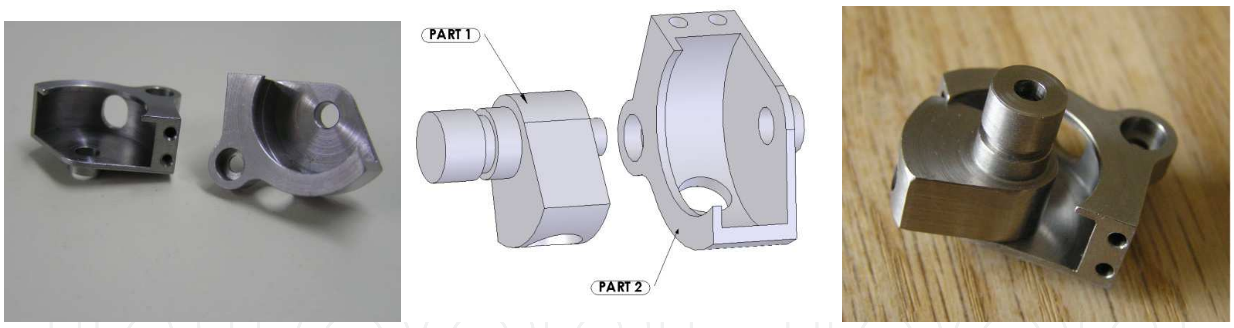

Fig. 7. RE dimensional tolerancing case study parts and assembly

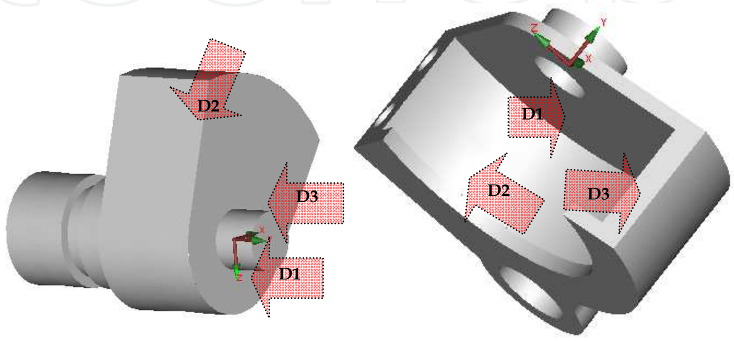

Fig. 8. Critical RE-features of the RE dimensional tolerancing case study parts

Hole	d_{M_h} **(mm)**	F_h **(mm)**	R_{a_h}
h #1	22.136	0.008	3.8
h #2	22.128	0.008	
h #3	22.091	0.003	
h #4	22.078	0.004	
U_h = 0.009mm			
Shaft	d_{M_s} **(mm)**	F_s **(mm)**	R_{a_s}
s #1	21.998	0.003	2.4
s #2	21.984	0.005	
s #3	21.979	0.006	
s #4	21.972	0.005	
U_s = 0.012mm			

Table 2. Input data related to case study RE-features

Suggested Fits	
∅ 22.000 H12 / g7, H12 / g8, H12 / g9, H12 / g10, H12 / g11, H12 / h8, H12 / h9, H12 / h10, H12 / h11 ∅ 22.050 H11 / e9, H11 / e10, H11 / e11, H11 / f10,	*∅22.000* D10 / h8, D11 / h8, D12 / h8, E11/ h8, E12/ h8, F11 / h8, F12 / h8, G11 / h8, G12 / h8, H12 / h8 D10 / h9, D11 / h9, D12 / h9, E11/ h8, E12/ h9, F11 / h9, F12 / h9, G11 / h9, G12 / h9, H12 / h9 D10 / h10, D11 / h10, D12 / h10, E11/ h10, E12/ h10, F11 / h10, F12 / h10, G11 / h10, G12 / h10, H12 / h10 D10 / h11, D11 / h11, D12 / h11, E11/ h11, E12/ h11, F11 / h11, F12 / h11, G11 / h11, G12 / h11, H12 / h11

H11 / f11, H11 / g7, H11 / g8, H11 / g9, H11 / g10, H11 / g11, H11 / h8, H11 / h9, H11 / h10, H11 / h11, H12 / e9, H12 / e10, H12 / e11, H12 / f10, H12 / f11, H12 / g7, H12 / g8, H12 / g9, H12 / g10, H12 / g11 H12 / h8, H12 / h9, H12 / h10, H12 / h11	*∅22.050* F10 / h10, F11 / h10, F12 / h10, G10 / h10, G11 / h10, G12 / h10, H11 / h10, H12 / h10 F10 / h11, F11 / h11, F12 / h11, G10 / h11, G11 / h11, G12 / h11, H11 / h11, H12 / h11
Preferred Fits	
∅ 22.050 H11 / e9, H11 / h8, H11 / h9, H11 / h11	∅ 22.000 D10 / h9, D11 / h9, D10 / h8, D11 / h8, D10 / h11, D11 / h11
∅ 22.000 D10 / h9, D11 / h9, D10 / h8, D11 / h8	

Table 3. Sets of Suggested and Preferred fits for the case study

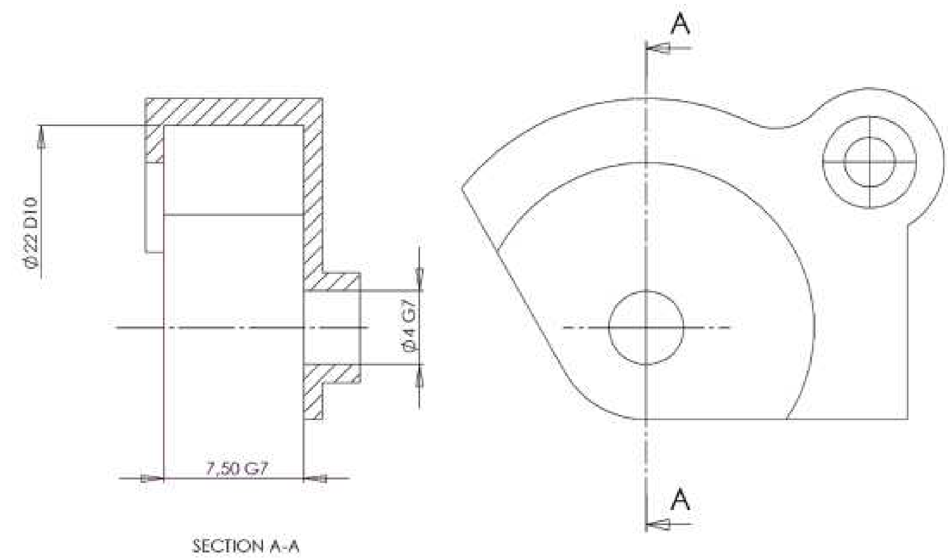

Fig. 9. Selected ISO 286 fits applied on the case study mechanical drawing

6.2 Application example of RE geometrical tolerancing

A new optical sensor made necessary the redesign of an existing bracket that had to be fastened on the old system through the original group of 4 x M5 threaded bolts, Figure 10. The accuracy of the location and orientation of the sensor bracket was considered critical for the sensor proper function. Bracket redesign had to be based on two available old and not used reference components. In the following, the method application is focused only on the allocation of position tolerances for the four bracket mounting holes. The problem represents, apparently, a typical fixed fastener case. For the chapter economy the input data and the produced results for the allocation of the position tolerance for the through hole H1, Figure 11, are only here presented and discussed. Standard diameter of the holes is 5.3mm ± 0.1mm and minimum clearance between hole and M5 screw 0.2mm.The weight coefficient in the relationships (21) was taken k=1.

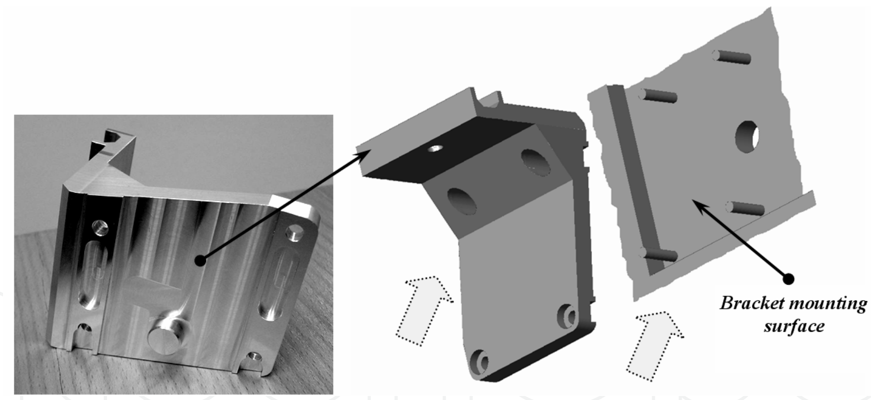

Fig. 10. Reference RE part and mating parts assembly

The CAD model of the redesigned new bracket was created in Solidworks taking into account the measured data of the reference components and, as well as, the new sensor mounting requirements. Datum and feature *(hole H1)* related input data are given in Table 4. Four candidate datum features A, B, C and D were considered, Figure 11. In step (a) 10 candidate DRFs (|A|B|C|, |A|C|B|, |A|B|D|, |A|D|B|, |A|C|D|, |A|D|C|, |D|A|C|, |D|A|B|, |D|B|A|, |D|C|A|) were produced by the algorithm for 7 position tolerance sizes, 0.06-0.18mm and consequently 10 sets of candidate theoretical dimensions, [$(C_P^{(ij)}X, C_P^{(ij)}Y)$, i=3,…9, j=1,…,10]. Negligible form and orientation deviations of datums A and D reduced the DRFs to the first six of them as |D|A|C|≡|A|D|C|≡|D|C|A| and |D|A|B|≡|A|D|B|≡|D|B|A|, having thus provided for 751 suggested tolerances in the following step (b). A representative sample of these tolerances is shown in Table 5. Although computational time difference is not significant, it is noticed that the quantity of the suggested results is strongly influenced by the number of the initially recognized possible datum features, weight coefficient k of the constraints (21), parameter δ of the equations (23) and the number of the available reference components.

d_M (mm)			**A|B|C**	**A|C|B**	**A|B|D**	**A|C|D**	**A|D|B**	**A|D|C**
Part1	5.291	X_{M1}	7.023	7.031	7.025	31.014	31.007	31.011
		Y_{M1}	5.972	5.961	50.981	5.964	50.980	50.978
Part2	5.244	X_{M2}	6.988	7.004	6.987	30.973	30.962	30.971
		Y_{M2}	6.036	6.019	51.028	6.017	51.026	51.012

Datum	***RF_{DF} (mm)***		***RO (mm)***		***RO_{DF} (mm)***	
	Part 1	***Part 2***	***Part 1***	***Part 2***	***Part 1***	***Part 2***
A	0.008	0.011	0.005	0.006	B-0.024 /C-0.021 /D-0.008	B-0.027/C-0.019 /D-0.007
B	0.026	0.023	0.012	0.010	A-0.041 /C-0.039 /D-0.035	A-0.046/C-0.043 /D-0.042
C	0.016	0.021	0.016	0.011	A-0.042 /B-0.034 /D-0.041	A-0.038/B-0.036 /D-0.033
D	0.005	0.008	0.003	0.007	A-0.010 /B-0.022 /C-0.020	A-0.009/ B-0.025 /C-0.017

Table 4. Input data related to case study feature and datum

<table>
<tr><th>T_{POS}</th><th>DRF</th><th>X</th><th>Y</th><th>Material Modifier</th><th>T_{POS}</th><th>DRF</th><th>X</th><th>Y</th><th>Material Modifier</th></tr>
<tr><td>∅ 0.060</td><td>|A|B|C|</td><td>7.000</td><td>6.000</td><td>MMC</td><td>∅ 0.120</td><td>|A|D|C|</td><td>30.950</td><td>50.950</td><td>MMC</td></tr>
<tr><td>∅ 0.060</td><td>|A|C|B|</td><td>6.950</td><td>5.950</td><td>LMC</td><td>∅ 0.120</td><td>|A|D|C|</td><td>30.900</td><td>51.000</td><td>LMC</td></tr>
<tr><td>∅ 0.060</td><td>|A|C|B|</td><td>7.100</td><td>6.000</td><td>LMC</td><td>...........</td><td></td><td></td><td></td><td></td></tr>
<tr><td>∅ 0.060</td><td>|A|B|D|</td><td>7.050</td><td>51.000</td><td>LMC</td><td>∅ 0.140</td><td>|A|B|C|</td><td>7.050</td><td>6.000</td><td>MMC</td></tr>
<tr><td>∅ 0.060</td><td>|A|C|D|</td><td>30.950</td><td>6.000</td><td>MMC</td><td>∅ 0.140</td><td>|A|C|B|</td><td>7.000</td><td>5.950</td><td>-</td></tr>
<tr><td>∅ 0.060</td><td>|A|D|B|</td><td>31.000</td><td>51.000</td><td>MMC</td><td>∅ 0.140</td><td>|A|C|B|</td><td>7.050</td><td>6.000</td><td>-</td></tr>
<tr><td>∅ 0.060</td><td>|A|D|B|</td><td>31.050</td><td>51.050</td><td>LMC</td><td>∅ 0.140</td><td>|A|B|D|</td><td>7.000</td><td>51.100</td><td>LMC</td></tr>
<tr><td>∅ 0.060</td><td>|A|D|C|</td><td>30.950</td><td>51.000</td><td>MMC</td><td>∅ 0.140</td><td>|A|C|D|</td><td>31.050</td><td>6.000</td><td>MMC</td></tr>
<tr><td>∅ 0.060</td><td>|A|D|C|</td><td>31.050</td><td>50.950</td><td>LMC</td><td>∅ 0.140</td><td>|A|C|D|</td><td>31.100</td><td>6.000</td><td>LMC</td></tr>
<tr><td>...........</td><td></td><td></td><td></td><td></td><td>∅ 0.140</td><td>|A|C|D|</td><td>31.100</td><td>6.050</td><td>LMC</td></tr>
<tr><td>∅ 0.080</td><td>|A|B|C|</td><td>7.000</td><td>6.000</td><td>-</td><td>∅ 0.140</td><td>|A|D|B|</td><td>30.950</td><td>51.000</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|B|C|</td><td>7.100</td><td>6.000</td><td>LMC</td><td>∅ 0.140</td><td>|A|D|B|</td><td>30.900</td><td>51.000</td><td>MMC</td></tr>
<tr><td>∅ 0.080</td><td>|A|C|B|</td><td>7.050</td><td>6.000</td><td>MMC</td><td>∅ 0.140</td><td>|A|D|C|</td><td>30.950</td><td>50.950</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|B|D|</td><td>7.000</td><td>51.000</td><td>-</td><td>∅ 0.140</td><td>|A|D|C|</td><td>30.950</td><td>51.000</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|B|D|</td><td>7.100</td><td>51.000</td><td>LMC</td><td>∅ 0.140</td><td>|A|D|C|</td><td>30.900</td><td>51.000</td><td>MMC</td></tr>
<tr><td>∅ 0.080</td><td>|A|C|D|</td><td>31.000</td><td>6.000</td><td>-</td><td>...........</td><td></td><td></td><td></td><td></td></tr>
<tr><td>∅ 0.080</td><td>|A|D|B|</td><td>31.000</td><td>51.000</td><td>MMC</td><td>∅ 0.160</td><td>|A|B|C|</td><td>6.950</td><td>5.950</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|D|B|</td><td>30.950</td><td>51.000</td><td>LMC</td><td>∅ 0.160</td><td>|A|B|C|</td><td>6.950</td><td>6.000</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|D|C|</td><td>31.000</td><td>6.000</td><td>-</td><td>∅ 0.160</td><td>|A|C|B|</td><td>7.000</td><td>5.950</td><td>-</td></tr>
<tr><td>∅ 0.080</td><td>|A|D|C|</td><td>30.095</td><td>51.050</td><td>MMC</td><td>∅ 0.160</td><td>|A|C|B|</td><td>7.150</td><td>6.000</td><td>LMC</td></tr>
<tr><td>∅ 0.080</td><td>|A|D|C|</td><td>31.050</td><td>51.050</td><td>LMC</td><td>∅ 0.160</td><td>|A|B|D|</td><td>7.000</td><td>51.100</td><td>MMC</td></tr>
<tr><td>...........</td><td></td><td></td><td></td><td></td><td>∅ 0.160</td><td>|A|B|D|</td><td>7.100</td><td>51.000</td><td>LMC</td></tr>
<tr><td>∅ 0.100</td><td>|A|B|C|</td><td>7.000</td><td>6.000</td><td>-</td><td>∅ 0.160</td><td>|A|C|D|</td><td>30.950</td><td>6.000</td><td>-</td></tr>
<tr><td>∅ 0.100</td><td>|A|C|B|</td><td>7.000</td><td>6.000</td><td>-</td><td>∅ 0.160</td><td>|A|C|D|</td><td>30.900</td><td>5.900</td><td>LMC</td></tr>
<tr><td>∅ 0.100</td><td>|A|C|B|</td><td>7.050</td><td>6.000</td><td>-</td><td>∅ 0.160</td><td>|A|D|B|</td><td>31.000</td><td>51.050</td><td>-</td></tr>
<tr><td>∅ 0.100</td><td>|A|B|D|</td><td>7.000</td><td>51.000</td><td>-</td><td>∅ 0.160</td><td>|A|D|C|</td><td>31.050</td><td>51.000</td><td>-</td></tr>
<tr><td>∅ 0.100</td><td>|A|B|D|</td><td>6.950</td><td>51.000</td><td>LMC</td><td>∅ 0.160</td><td>|A|D|C|</td><td>30.900</td><td>51.000</td><td>MMC</td></tr>
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<tr><td>∅ 0.100</td><td>|A|C|D|</td><td>30.950</td><td>5.950</td><td>MMC</td><td>...........</td><td></td><td></td><td></td><td></td></tr>
<tr><td>∅ 0.100</td><td>|A|D|B|</td><td>31.000</td><td>51.050</td><td>MMC</td><td>∅ 0.180</td><td>|A|B|C|</td><td>7.050</td><td>6.000</td><td>-</td></tr>
<tr><td>∅ 0.100</td><td>|A|D|B|</td><td>30.950</td><td>50.900</td><td>LMC</td><td>∅ 0.180</td><td>|A|B|C|</td><td>7.150</td><td>6.000</td><td>LMC</td></tr>
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<tr><td>...........</td><td></td><td></td><td></td><td></td><td>∅ 0.180</td><td>|A|B|D|</td><td>6.900</td><td>51.000</td><td>MMC</td></tr>
<tr><td>∅ 0.120</td><td>|A|B|C|</td><td>7.050</td><td>6.050</td><td>MMC</td><td>∅ 0.180</td><td>|A|B|D|</td><td>6.900</td><td>51.050</td><td>MMC</td></tr>
<tr><td>∅ 0.120</td><td>|A|B|C|</td><td>7.100</td><td>6.000</td><td>LMC</td><td>∅ 0.180</td><td>|A|C|D|</td><td>31.050</td><td>6.000</td><td>-</td></tr>
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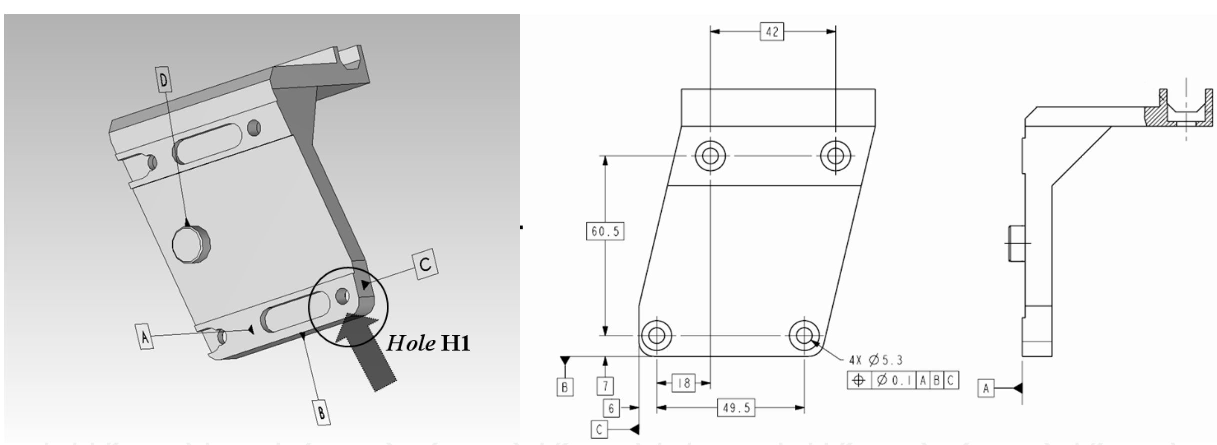
<tr><td>∅ 0.120</td><td>|A|D|B|</td><td>30.900</td><td>50.950</td><td>LMC</td><td>∅ 0.180</td><td>|A|D|C|</td><td>30.900</td><td>50.900</td><td>LMC</td></tr>
</table>

Table 5. Representative sample of application example suggested position tolerances

T_{POS}	*DRF*	*X*	*Y*	*Material Modifier*	T_{POS}	*DRF*	*X*	*Y*	*Material Modifier*
∅ 0.080	|A|B|C|	7.000	6.000	-	∅ 0.100	|A|C|B|	7.000	6.000	-
∅ 0.800	|A|B|D|	7.000	51.000	-	∅ 0.100	|A|C|B|	7.050	6.000	-
∅ 0.080	|A|C|D|	31.000	6.000	-	∅ 0.100	|A|B|D|	7.000	51.000	-
∅0.080	|A|D|C|	31.000	51.000	-	∅ 0.100	|A|C|D|	31.000	6.000	-
∅ 0.100	|A|B|C|	7.000	6.000	-	∅ 0.100	|A|D|B|	31.000	51.000	-
					∅ 0.100	|A|D|C|	31.000	51.000	-

Table 6. Set of preferred position tolerances for the application example

The preferred 11 out of 751 position tolerances of the Table 6 were obtained applying the selection options and guidelines of the section 4.4. Parallel results were obtained for the other three holes. As it came out, all of them belong to a group of holes with common DRF. The position tolerance size ∅ 0.100mm and the DRF |A|B|C| were finally chosen by the machine shop. Theoretical hole location dimensions are shown in Figure 11. The results were experimentally verified and approved. Time needed for the entire task was 12min (CMM) + 6min (Analysis) =18min. The usual trial-and-error way would, apparently, require considerably longer time and produce doubtful results. Reliability of the results can certainly be affected by failing to recognize initial datum features. In machine shop practice however, risk for something like that is essentially negligible.



(a) (b)

Fig. 11. Case study datums (a) and the selected position tolerance of the case study (b)

6.3 Application example of cost-effective RE tolerancing

In the assembly of components A-B-C of Figure 12 the dimension D4 = 74.95 ± 0.25mm is controlled through the dimensional chain,

$$D_4 = \sin D_3 \left(D_1 + D_2 + D_7 + D_{10} - D_{11} - D_5 - D_6 + D_9\right)$$

with D_1 = 190mm, D_2 = 15mm, D_3 = 45°, D_5 = 14mm, D_6 = 95mm, D_7 = 20mm, D_9 = 75mm, D_{10} = 12mm, D_{11} = 97mm. Component B is reverse engineered and needs to be remanufactured with main intention to fit and perform well in the existing assembly. All of the original component design and manufacturing information is, however, not available and the dimensional accuracy specifications for component B reconstruction have to be reestablished.

The machine shop where the part will be manufactured has an IT6 best capability and its DFF processed and the results stored. Alternatives for parts A, C and B, provided by the RE dimensional analysis of Section 3, are shown in Table 7(a) and (b) respectively. The 64 possible combinations of the part B alternatives are filtered out according to the tolerance chain constrains and, as a result, 24 combinations occur for the dimensions and tolerances $D_5 \pm t_5$, $D_6 \pm t_6$, $D_7 \pm t_7$ as shown in Table 8. The optimum combination that corresponds to the minimum accuracy cost is the combination 64.

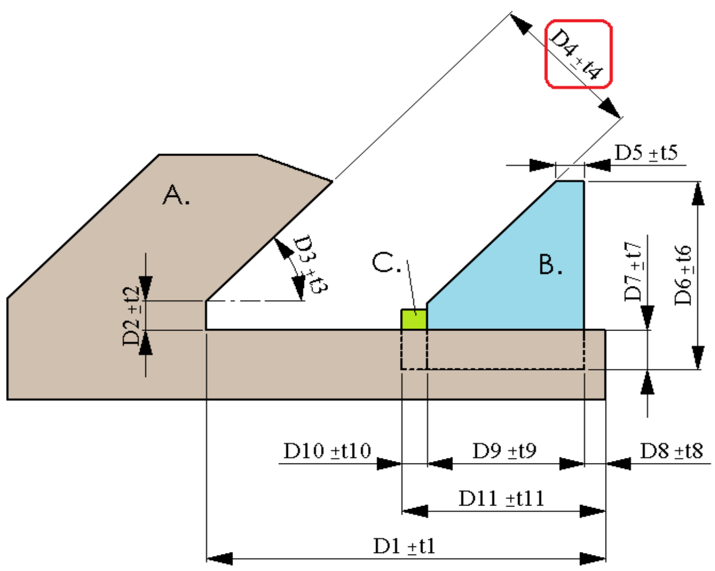


Fig. 12. Application example of Cost-Effective RE tolerancing

	Alternative 1		*Alternative 2*	
	D_{min}	D_{max}	D_{min}	D_{max}
D_1	190.01	190.04	189.97	190.00
D_2	14.99	15.02	14.98	15.00
D_3	45.01	45.02	44.98	45.00
D_7	20.00	20.01	19.99	20.005
D_{10}	12.01	12.025	11.98	12.00
D_{11}	97.00	97.02	96.98	97.025

a/a	D_5		D_6		D_9	
	D_{min}	D_{max}	D_{min}	D_{max}	D_{min}	D_{max}
1	14.00	14.12	95.03	95.15	74.67	75.23
2	13.89	14.30	94.98	95.11	75.00	75.01
3	13.98	13.99	94.86	94.99	74.77	74.99
4	13.89	14.01	94.95	95.14	74.98	75.10

Table 7. Dimensional alternatives for parts A, C and B

Combination	D_5		D_6		D_9		Accuracy Cost
	D_{min}	D_{max}	D_{min}	D_{max}	D_{max}	D_{min}	
2	14.00	14.12	95.03	95.15	75.00	75.01	3.3540
4	14.00	14.12	95.03	95.15	74.98	75.10	0.9035
6	14.00	14.12	94.98	95.11	75.00	75.01	3.3190
8	14.00	14.12	94.98	95.11	74.98	75.10	0.8684
10	14.00	14.12	94.86	94.99	75.00	75.01	3.3190
12	14.00	14.12	94.86	94.99	74.98	75.10	0.8684
14	14.00	14.12	94.95	95.14	75.00	75.01	3.1836
34	13.98	13.99	95.03	95.15	75.00	75.01	4.6426
36	13.98	13.99	95.03	95.15	74.98	75.10	2.1920
38	13.98	13.99	94.98	95.11	75.00	75.01	4.6075
40	13.98	13.99	94.98	95.11	74.98	75.10	2.1569
42	13.98	13.99	94.86	94.99	75.00	75.01	4.6075
43	13.98	13.99	94.86	94.99	74.77	74.99	2.0459
44	13.98	13.99	94.86	94.99	74.98	75.10	2.1569
46	13.98	13.99	94.95	95.14	75.00	75.01	4.4721
48	13.98	13.99	94.95	95.14	74.98	75.10	2.0216
50	13.89	14.01	95.03	95.15	75.00	75.01	3.3540
52	13.89	14.01	95.03	95.15	74.98	75.10	0.9035
54	13.89	14.01	94.98	95.11	75.00	75.01	3.3190
56	13.89	14.01	94.98	95.11	74.98	75.10	0.8684
58	13.89	14.01	94.86	94.99	75.00	75.01	3.3190
59	13.89	14.01	94.86	94.99	74.77	74.99	0.7574
62	13.89	14.01	94.95	95.14	75.00	75.01	3.1836
64	13.89	14.01	94.95	95.14	74.98	75.10	0.7330

Table 8. Filtered out combinations

7. Conclusion

In industrial manufacturing, tolerance assignment is one of the key activities in the product creation process. However, tolerancing is much more difficult to be successfully handled in RE. In this case all or almost all of the original component design and manufacturing information is not available and the dimensional and geometric accuracy specifications for component reconstruction have to be re-established, one way or the other, practically from scratch. RE-tolerancing includes a wide range of frequently met industrial manufacturing problems and is a task that requires increased effort, cost and time, whereas the results, usually obtained by trial-and-error, may well be not the best. The proposed methodology offers a systematic solution for this problem in reasonable computing time and provides realistic and industry approved results. This research work further extends the published research on this area by focusing on type of tolerances that are widely used in industry and almost always present in reverse engineering applications. The approach, to the extent of the author's knowledge, is the first of the kind for this type of RE problems that can be directly implemented within a CAD environment. It can also be considered as a pilot for further research and development in the area of RE tolerancing. Future work is oriented towards the computational implementation of the methodology in 3D-CAD environment, the RE *composite* position tolerancing that concerns patterns of repetitive features, the methodology application on the whole range of GD&T types and the integration of function oriented wear simulation models in order to evaluate input data that come from RE parts that bear considerable amount of wear.

8. References

Abella, R.J.; Dashbach, J.M. &, McNichols, R.J. (1994). Reverse Engineering Industrial Applications, *Computers & Industrial Engineering*, Vol.26, No. 2, pp. 381-385, ISSN: 0360-8352

Anselmetti, B. & Louati, H. (2005). Generation of manufacturing tolerancing with ISO standards, *International Journal of Machine Tools & Manufacture,* Vol. 45, pp. 1124–1131, ISSN: 0890-6955

ASME Standard. (1994). *Y14.5.1M-1994: Mathematical Definition of Dimensioning and Tolerancing Principles*, The American Society of Mechanical Engineers, New York, USA

ASME Standard. (2009). *Y14.5M–2009: Dimensioning and Tolerancing*, The American Society of Mechanical Engineers, New York, USA

Bagci, E. (2009). Reverse engineering applications for recovery of broken or worn parts and re-manufacturing: Three case studies, *Advances in Engineering Software*, Vol. 40, pp. 407–418, ISSN: 0965-9978

Borja, V., Harding, J.A. & Bell, B. (2001). A conceptual view on data-model driven reverse engineering, *Int. J. Prod. Res.,* Vol.39, No 4, pp. 667-687

Chant, A.; Wilcock, D. & Costello, D. (1998). The Determination of IC engine inlet port geometries by Reverse Engineering, *International Journal of Advanced Manufacturing Technology,* Vol.14, pp. 65–69, ISSN: 0268-3768

Chen, L.C. & Lin G.C.I. (2000). Reverse engineering in the design of turbine blades: a case study in applying the MAMDP, *Robotics and Computer Integrated Manufacturing*, Vol.16, pp. 161-167, ISSN: 0736-5845

Dan, J. & Lancheng, W. (2006). Direct generation of die surfaces from measured data points based on springback compensation, *International Journal of Advanced Manufacturing Technology*, Vol.31, pp. 574–579, ISSN: 0268-3768

Dimitrellou S.; Kaisarlis J. G.; Diplaris C. S. & Sfantsikopoulos M. M. (2009). A Cost Optimal Tolerance Synthesis Methodology for Design and Manufacturing Integration, *Proceedings of 11th CIRP Conference on Computer Aided Tolerancing*, Annecy, France March 26 – 27, 2009

Dimitrellou, S.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2007b). A Systematic Approach for Cost Optimal Tolerance Design, *Proceedings of the International Conference On Engineering Design - ICED'07*, IBSN: 1-904670-02-4, Paris, France, August 28 – 31, 2007

Dimitrellou, S.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2007a). Cost-competent Tolerancing in CAD, *International Journal of Advanced Manufacturing Technology*, Vol. 35, Nos. 5-6, pp. 519-526, ISSN: 0268-3768

Dimitrellou, S.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2008). Tolerance Elements. An Alternative Approach for Cost Optimum Tolerance Transfer, *Journal of Engineering Design*, *Special Issue: Cost Engineering,* Vol. 19, No. 2, pp. 173-184, ISSN: 0954-4828

Dimitrellou, S.; Kaisarlis, G.J.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2007c). A systematic approach for functional and cost optimal dimensional tolerancing in reverse engineering, *Proceedings of XVII Metrology and Metrology Assurance (MMA 2007) Conference*, ISBN: 9789543340613, pp. 193 – 197, Sozopol, Bulgaria, September 10 - 14, 2007

Diplaris, S. & Sfantsikopoulos, M.M. (2006). Maximum material condition in process planning, *Production Planning & Control*, Vol. 17, No. 3, pp. 293-300, ISSN: 0953-7287

Drake, P.J. (1999). *Dimensioning and Tolerancing Handbook*, ISBN 0-07-018131-4, McGraw – Hill, New York, USA

Endo, M. (2005). Reverse Engineering and CAE, *JSME International Journal, Series:C*, Vol.48, No. 2, pp. 218-223

Fischer, B.R. (2004). *Mechanical Tolerance Stackup and Analysis.* Marcel Dekker, ISBN: 0824753798, New York, USA

Flack, D. (2001). *CMM Measurement Strategies NPL Measurement Good Practice Guide No. 41*. NPL - Crown, ISSN 1368-6550, Teddington, UK.

Ingle, K.A. (1994). *Reverse engineering,* McGraw Hill, New York, USA.

ISO Standard, (2004). *ISO-1101: Geometrical Product Specifications (GPS) - Geometrical tolerancing - Tolerances of form, orientation, location and run-out*, The International Organization for Standardization, Geneva, Switzerland

ISO Standard, (1998). *ISO 5458: Geometrical Product Specifications (GPS) - Geometrical Tolerance - Position Tolerancing*, The International Organization for Standardization, Geneva, Switzerland

ISO Standard, (1988). *ISO 2692: Technical drawings - geometrical tolerancing - maximum material principle*, The International Organization for Standardization, Geneva, Switzerland

Kaisarlis, G.J.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2000). A Knowledge-Based System for Tolerance Allocation in Reverse Engineering, *Proceedings of 33rd MATADOR Conference*, D.R. Hayhurst (Ed), pp. 527 - 532, Manchester, UK, July 13-14, 2000, Springer Verlag, IBSN: 1852333235, London, UK

Kaisarlis, G.J.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2004). Datum Identification in Reverse Engineering, *Proceedings of IFAC Conference on Manufacturing, Modelling, Management and Control*, G. Chryssolouris, D. Mourtzis (Ed), pp. 179 – 183, Athens, Greece, October 21-22, 2004, Elsevier Science Ltd, , ISBN: 0080445624, Amsterdam, The Netherlands

Kaisarlis, G.J.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2007). Position Tolerancing in Reverse Engineering: The Fixed Fastener Case, *Journal of Engineering Manufacture, Proceedings of the Institution of Mechanical Engineers, Part B,* Vol. 221, pp. 457 - 465, ISSN: 0954-4054

Kaisarlis, G.J.; Diplaris, S.C. & Sfantsikopoulos, M.M. (2008). Geometrical position tolerance assignment in reverse engineering, *International Journal of Computer Integrated Manufacturing*, Vol. 21, No 1, pp. 89 - 96, ISSN: 0951-192X

Martin, P.; Dantan, J.Y. & D'Acunto, A. (2011). Virtual manufacturing: prediction of work piece geometric quality by considering machine and set-up accuracy, *International Journal of Computer Integrated Manufacturing,* Vol.24, No. 7, pp. 610-626, ISSN: 0951-192X

Mavromihales, M.; Mason, J. & Weston, W. (2003). A case of Reverse Engineering for the manufacture of wide chord fan blades (WCFB) in Rolls Royce aero engines, *Journal of Materials Processing Technology*, Vol.134, pp. 279-286, ISSN: 0924-0136

Raja, V. & Fernandes, K. (2008). *Reverse Engineering : an industrial perspective*, Springer Series in Advanced Manufacturing, Springer-Verlag, ISBN 978-1-84628-855-5, London, UK

Singh, P.K.; Jain, P.K.; & Jain, S.C. (2009a). Important issues in tolerance design of mechanical assemblies. Part 2: Tolerance Synthesis. *Proc. IMechE, Part B: J. Engineering Manufacture,* Vol.223, pp. 1225-1247, ISSN: 0954-4054

Singh, P.K.; Jain, P.K.; & Jain, S.C. (2009b). Important issues in tolerance design of mechanical assemblies. Part 2: Tolerance Analysis. *Proc. IMechE, Part B: J. Engineering Manufacture*, Vol.223, pp. 1249-1287, ISSN: 0954-4054

Thompson, W.B.; Owen, J.C.; de St. Germain, H.J.; Stark Jr, S.R. & Henderson, T.C. (1999). Feature based reverse engineering of mechanical parts, *IEEE Transactions on Robotics and Automation*, Vol.15, pp. 56-66, ISSN: 1042-296X

Várady, T.; Martin, R. & Cox, J. (1997). Reverse engineering of geometric models—an introduction, *Computer-Aided Design*, Vol. 29, No 4, pp. 255-268, ISSN: 0010-4485

VPERI. (2003). *Army research office virtual parts engineering research initiative*, 20.09.2011, Available from: <http://www.cs.utah.edu/gdc/Viper/Collaborations/VPERI-Final-Report.pdf>

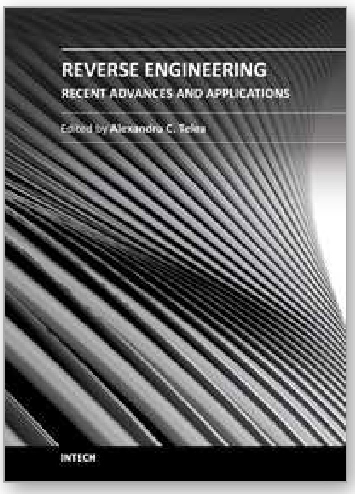
Wego, W. (2011). *Reverse engineering: Technology of Reinvention*, CRC Press – Taylor & Francis, ISBN 978-1-4398-0631-9, Florida, USA

Yau, H.T. (1997). Reverse engineering of engine intake ports by digitization and surface approximation, *Int. Journal of Machine Tools & Manufacturing*, Vol.37, No. 6, pp. 855-871, ISSN: 0890-6955

Yuan, X.; Zhenrong, X. & Haibin, W. (2001). Research on integrated RE technology for forming sheet metal with a free form surface, *Journal of Materials Processing Technology*, Vol.112, pp. 153–156, ISSN: 0924-0136

Zhang, W. (2003) Research into the engineering application of Reverse Engineering technology, *Journal of Materials Processing Technology*, Vol. 139, pp. 472-475, ISSN: 0924-0136

Zheng, J.; Li, Z. & Chen, X. (2004). Broken area modeling for repair automation, *Journal of Advanced Manufacturing Systems*, Vol. 3, No. 2, pp. 129-140, ISSN: 0219-6867



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Reverse engineering encompasses a wide spectrum of activities aimed at extracting information on the function, structure, and behavior of man-made or natural artifacts. Increases in data sources, processing power, and improved data mining and processing algorithms have opened new fields of application for reverse engineering. In this book, we present twelve applications of reverse engineering in the software engineering, shape engineering, and medical and life sciences application domains. The book can serve as a guideline to practitioners in the above fields to the state-of-the-art in reverse engineering techniques, tools, and use-cases, as well as an overview of open challenges for reverse engineering researchers.

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InTech Europe
University Campus STeP Ri
Slavka Krautzeka 83/A
51000 Rijeka, Croatia
Phone: +385 (51) 770 447
Fax: +385 (51) 686 166
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InTech China
Unit 405, Office Block, Hotel Equatorial Shanghai
No.65, Yan An Road (West), Shanghai, 200040, China
中国上海市延安西路65号上海国际贵都大饭店办公楼405单元
Phone: +86-21-62489820
Fax: +86-21-62489821