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The Calculation Method of the Inspection Tolerance Zone When Two Datum Features Apply MMC/LMC

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Abstract

When the datum features apply the maximum or least material condition (MMC/LMC), the shift tolerance, because of the departure of the actual mating size of the datum features from their extreme condition, can be added to the stated tolerance of the related feature, thereby improving the acceptance rate of the parts. However, it is difficult to calculate the shift tolerance when more than one datum feature applies MMC/LMC. In this paper, a general calculation method of the inspection tolerance zone (ITZ) based on the linkage mechanism model is proposed. The Datum Feature Simulators (DFSs) at actual virtual condition are used to construct the measurement datum reference frame (DRF), and the DFSs at the stated MMC/LMC condition are used to construct the design DRF. In the proposed methodology, the relative position between the design DRF and the measurement DRF is simulated by the series combination of two linkage mechanisms, the measurement DRF is fixed at the mechanism frame, the design DRF is fixed at target linkage, and the kinematic envelope of the design tolerance zone at design DRF corresponds to the ITZ. The combination of the datum features for applying MMC/LMC is investigated, and the combination principles of the representation mechanism for different combinations of the datum features are discussed. An example is given to demonstrate the proposed method.

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Keywords: Inspection tolerance zone; MMC/LMC; shift tolerance; representation mechanism

1. Introduction

Theoretically, the maximum or least material condition (MMC/LMC) may be used in assembly tolerance specification to great advantages. The MMC/LMC allows an inspector to increase the geometric tolerance when a real feature is manufactured away from its MMC/LMC size. However, the use of MMC/LMC is difficult for many designers, and the verification of these specification is still unable for many metrology computer program, especially, for the specification that multiple datum features apply MMC/LMC. The main reason is probably the difficulty in calculation of the shift tolerance when the real datum feature deviates from the extreme conditions. Therefore, the establishment of a mathematical model to calculate the shift tolerance is very important for the application of MMC/LMC.

For the tolerancing technology about the application of MMC/LMC, many research studies have been performed. ASME 14.5.1 [1] proposed a mathematical representation of the geometrical dimensioning and tolerancing. To define the tolerance specification at MMC/LMC, the notion of virtual boundary is presented in the ISO 2692:2014 standard [2]; this virtual boundary is used as virtual gauges since the early 1990s by [3]. Functional gauging techniques are typically used in industrial applications to control the conformance of simple MMC tolerances. However, the functional gauging techniques is unable to control the conformance of the tolerance feature with the definitions when more than one datum feature apply MMC. Moreover, it is unable to control the conformance of the tolerance feature at LMC, regardless of whether one datum feature or multiple datum features applying LMC. Anselmetti [4] illustrated the translation of the functional requirements of an elementary mechanism in geometrical specifications. The assembly requirement and the precision requirement are analyzed to define the maximum material requirement and the least material requirement using a virtual boundary, the location tolerance and a projected tolerance zone. Robinson [5] proposed the application of a maximum material part

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tolerancing principle. To integrate and address the geometrical tolerances, Bennis et al. presented a kinematic model in the geometric tolerance transfer problem [6], and for the inspection of parts defined with maximum material condition modifiers [7]. Later, the model was used in an assembly analysis based on the computation of the virtual and the resultant parts [8]. However, the definition of the virtual and resultant part is difficult to understand, and the calculation of the parameter of the kinematics model is not clear. Pairel et al. exposed a conceptual model of "virtual fitting gauges" [9]. Dantan at al. defined the gauge with internal mobilities for the assembly tolerance specification [10], and used virtual gauges with internal mobilities to verify the maximum material and least material requirements [11]. Shen et al. proposed a simulation-based approach to the tolerance and ease of assembly analyses for assemblies with pin/hole floating mating conditions [12] and for 3D slot features and tab features [13]. The algorithms developed account not only for bonus/shift tolerances but also for the feasibility of assembly. In [14], the extension of material conditions on complex surfaces and the definition of a new association criterion were developed, in order to specify hybrid prismatic surfaces with a surface contact zone and a zone with clearance. Anselmetti [15] proposed two complementary writings and several explanations for the application of the material requirements. The new description will improve the readability of the tolerance specification involved in the material condition and to facilitate the decoding of CAT software.

In this paper, a complete calculation methodology for the ITZ of true position of the related feature, when two datum features apply the MMC/LMC, is proposed. First, the calculation mechanism is introduced, and two Datum Feature Simulators (DFSs) are established for both the actual virtual condition and the extremal condition of the datum feature, which are design DFS (D_DFS) and measurement DFS (M_DFS), respectively, and the corresponding design DRF and measurement DRF are established according to D_DFSs and M_DFSs. Next, the configuration of the datum feature is investigated and all combinations of the datum feature that should apply MMC/LMC are enumerated according to the geometry of both related feature and datum features. Third, the different mechanisms are used to represent the relative motion between D_DFS and M_DFS, and the envelope of the tolerance zone on the measurement DRF is obtained based on the kinematical equation of the representation mechanism model; the envelope is the ITZ.

2. The calculation method of ITZ when two datum features apply the MMC/LMC

2.1. The inspection mechanism of the geometric tolerance

According to ASME Standard Y14.5 [16], when the datum feature applies the MMC/LMC, the stated geometric tolerance value applies when the datum features are at their extreme boundaries, i.e., maximum material boundary (MMB), maximum material virtual boundary (MMVB) or least material boundary (LMB), least material boundary (LMVB). In this paper, for the DFSs at MMB (MMVB) or LMB (LMVB) are used to establish the design DRF, the DFS is called D_DFS. In the measurement of the tolerance feature, the location elements of the measurement instrument or equipment are in contact with the actual boundary of the datum features. The locators are used to simulate the DFS of the actual datum feature and to establish the measurement DRF of the tolerance feature, these simulated DFSs are called M_DFS. Because of the deviation of the actual datum feature from the extreme boundary, the D_DFS of the datum feature is not the same as the corresponding M_DFS. Therefore, the design DRF has multiple candidate positions for a given measurement DRF, which will generate the shift tolerance. In this paper, a true position tolerance specification shown in Fig. 1 is used to illustrate the formation mechanism and the calculation method of the shift tolerance.

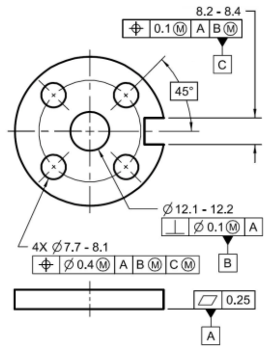


Fig.1 The example of the tolerance specification when the datum feature applying MMC

In this example, the true position of the hole pattern (4×Ø7.7-8.5 mm) has three datum features (bottom plane A, central hole B and slot C); central hole B and slot C have material modifier MMC. According to the definitions of the tolerance standard, these three datum features for the setting of the true position Ø0.4 mm of the hole pattern are the perfect plane, the perfect cylinder with a diameter Ø12 mm and perfect slot with a width 8.1 mm. The cylinder is perpendicular to the plane, and the central plane of the slot is perpendicular to the plane and contains the axis of the cylinder. The three perfect geometries are three D_DFSs to build the design DRF of the hole pattern. In the measurement of the actual position of the hole pattern, three locators are three M_DFSs to build the measurement DRF.

Because the part is manufactured away from its MMC size, the diameter and width of the M_DFS of both central hole B and slot C are always greater than 12 mm and 8.1 mm, respectively. The size difference of M_DFS and D_DFS causes the noncoincidence between the design DRF and the measurement DRF of the related feature. For a given measurement DRF, there are multiple candidate design DRFs, which explains why the measurement of the related feature is difficult when their datum features apply MMC. Furthermore, when the datum feature is at LMC, the size relationship between M_DFS and D_DFS of the corresponding datum features is contrary to that the datum features at MMC, but the measurement difficulty is the same. This problem still exists for the contact measurement or non-contact measurement.

In this case, the position and size relationship between M_DFS and D_DFS of the central hole B and the slot C can be described by Fig. 2. The cylinder and the cube with blue top surface denote the two D_DFSs of both central hole B and slot C, and the relative position between two D_DFSs is unchanged; therefore, they constitute a D_DFS component. The cylinder hole and slot denote the boundary of the related actual mating envelope of both central hole B and slot C, and their inverse geometric forms are the two M_DFSs of the central hole B and slot C, which constitute a M_DFS component. Suppose that the D_DFS component be a solid and the M_DFS component be a cavity; the D_DFS component can freely move in the M_DFS component. According to the definitions of the tolerance standard, the tolerance zone (the cylinder with Ø0.4 mm) of the hole pattern is defined coaxially with the z-axis of the design DRF, but the actual axis of the hole pattern is measured at the measurement DRF and has deviated from the z-axis of the measurement DRF. When the deviation is greater than 0.2 mm (the radius of the tolerance zone), if one moves the D_DFS component in the M_DFS component and makes the design tolerance zone inscribe the measured axis of the hole pattern, the position error of this hole pattern is still allowed. Thus, when we make the maximum relative motion of the D_DFS component with respect to the M_DFS component, the envelope of tolerance zone in the measurement DRF is the ITZ of the hole pattern. Certainly, the ITZ is greater than the design tolerance zone, and it is the contribution of the shift tolerance.

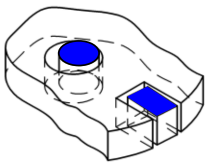


Fig. 2 The extreme condition and the actual condition of the datum feature

2.2. The representation of the maximum relative motion of D_DFS component with respect to M_DFS component

To describe the relative motion of the D_DFS component with respect to the M_DFS component completely, a relative motion relationship must be established. This kinematic equation is complicated, because it includes many factors, such as the geometry, the position and the size of all datum features. However, for the common datum geometry, such as the case shown in Fig. 1, the kinematic equation can be simplified. In the case shown in Fig. 1, both central hole B and slot C are perpendicular to the plane A; therefore, the relative motion of the D_DFS component with respect to the M_DFS component is a planar motion on the plane parallel to the plane A. To establish the kinematic equitation, both D_DFS and M_DFS components can be projected onto plane A, and the relative position of the D_DFS component with respect to the M_DFS component is transformed to the positional relationship between two groups of planar geometries. Fig. 3 shows the projection diagram of both D_DFS and M_DFS components on plane A; in this figure, the D_DFS component is a circle and a box in hidden line, and the M_DFS component is a circle and a box in solid line.

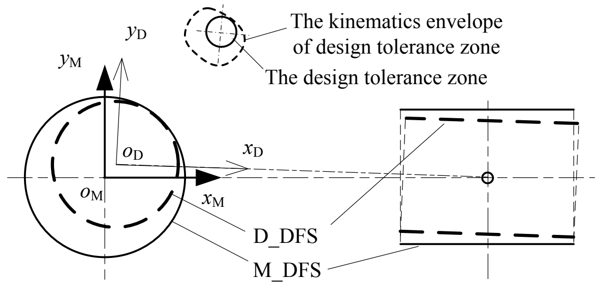


Fig.3 The relative movement between D_DFS component and M_DFS component

On the projection plane, the maximum motion of D_DFS component with respect to M_DFS component can be represented by a series combination of two simple linkage mechanisms. Fig. 4 is the diagram of two linkage mechanisms. The first mechanism is a fundamental mechanism consists of a crank guide-bar mechanism; the crank is a centerline of the hidden circle and the solid circle in Fig. 3, and the guide-bar goes through the center of the solid box in Fig.3. The second one is an oscillating-bar connected to the guide-bar of the fundamental mechanism by a pivot; that reciprocating oscillates relative to the guide-bar. The measurement DRF x_M-o_M-y_M is located at the frame of the fundamental mechanism; the origin o_M is the fixed pivot of the crank, and the x_M axis is collinear with the frame of the fundamental mechanism. The design DRF x_D-o_D-y_D is located at the oscillating-bar mechanism; the origin o_D is the oscillating center of the oscillating-bar, and the x_D axis is collinear with the oscillating-bar. The rotation of the crank and the oscillating of the oscillating-bar are two independent motions of the representation mechanism. The crank length is determined by the diameter difference between the solid circle and the hidden circle in Fig. 3, and the angular range of the oscillating-bar at any position of fundamental mechanism is determined by the parameter w, where w is the width difference between the solid box and the hidden box in Fig. 3. When the representation mechanism has moved in a complete cycle, the kinematics locus of the design tolerance zone (a circle with Ø0.4 mm) on the design DRF will envelope out an area on the measurement DRF, i.e. the ITZ. Therefore, the conclusion can be drawn from the above description that the representation mechanism can describe the maximum relative motion of the D_DFS component with respect to the M_DFS component, the crank length and the oscillating angle of the oscillating-bar can be used to calculate the ITZ.

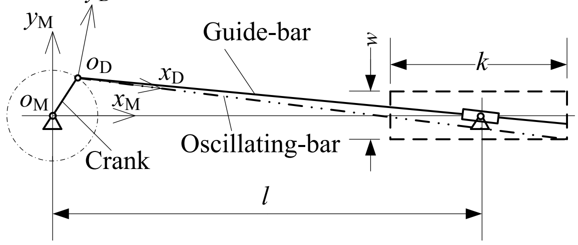


Fig. 4 The representation mechanism

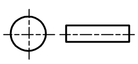
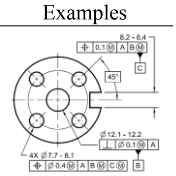
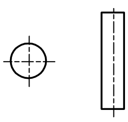
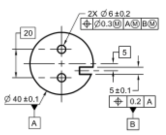
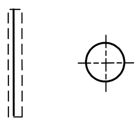
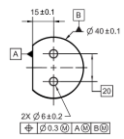
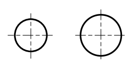
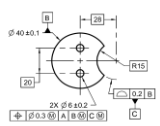
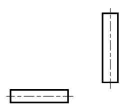
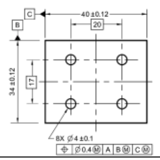
3. The calculation method of the ITZ for different datum features applying MMC/LMC

The determination method of the ITZ introduced above can be expanded to the situation that the datum feature has various geometries and combinations. In this section, we will give the representation mechanism for different datum systems.

3.1. The combination of the datum features when they apply MMC/LMC

To obtain the shift tolerance by applying MMC/LMC, the datum feature should be parallel to the centerline or center plane of the related feature. According to the analysis of the DOF of the geometric feature, the number of the feature of size that are parallel to the related feature should less than or equal to 2, for a given datum system of a related feature. Therefore, the collections of such datum feature combinations are enumerated easily. Tab. 1 is the list of the datum combination. According to tab. 1, there exist a planar datum feature that does not apply MMC/LMC, and it is perpendicular to both datum features applying MMC/LMC and the related feature; therefore, the relative position among the datum features can be simplified by that of 2D geometry. As a result, we can use the similar representation mechanism to represent the relative motion of the design DRF with respect to the measurement DRF.

Table 1 Combination of two datum features applying MMC/LMC

No.	Datum combination	Descriptions	Examples
1		The diameter feature and width feature, located in alignment with each other	
2		The diameter feature and width feature, located perpendicular to each other	
3		The diameter feature and width feature, located parallel to each other	
4		Two diameter features, located parallel to each other	
5		Two width features, located perpendicular to each other	

3.2. The calculation method of the ITZ for different combinations of the datum features applying MMC/LMC

According to MMC/LMC, when the datum feature is an external feature and apply MMC, or when the datum feature is an internal feature and apply LMC, the D_DFS of a datum feature is larger than its M_DFS. In contrast, when the datum feature is an internal and apply MMC, or when the datum feature is an external feature and apply LMC, the D_DFS of a datum feature is smaller than its M_DFS.

The relative position of the design DRF with respect to the measurement DRF can be represented by using the relative inclusion relation between the D_DFS component and the M_DFS component. When the size of D_DFS is larger than the size of M_DFS, the D_DFS component includes the M_DFS component; else, the D_DFS component is included by the M_DFS component. Regardless of the inclusion relationship, the calculation method of the ITZ is the same.

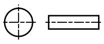
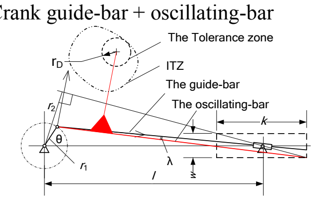
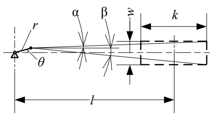
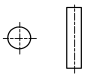
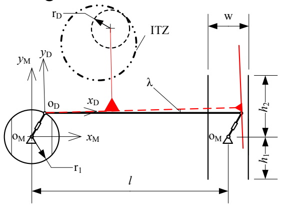
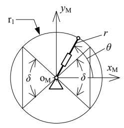
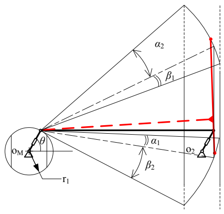
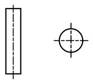
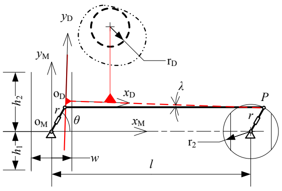
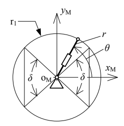
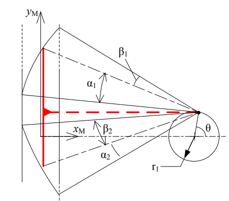
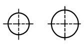
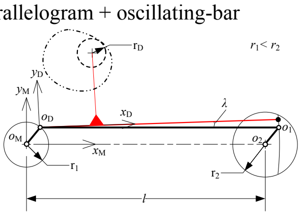
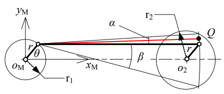
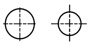
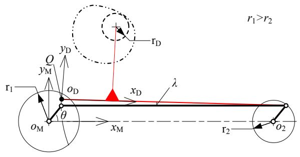
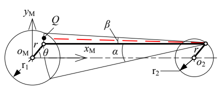
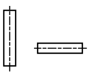
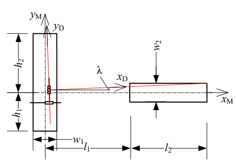
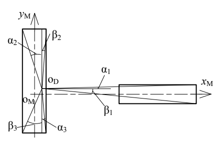
When the geometry of the datum feature is limited to the diameter feature and width feature, the D_DFS and M_DFS of the datum feature are a cylinder and prism, respectively. For the combination of the datum feature listed in Tab. 1, the projection of the D_DFS and the M_DFS on another planar datum are a circle and box, respectively. Therefore, the inclusion relationship among D_DFS and M_DFS is the inclusion relationship between two circles or two boxes.

The relative position of the D_DFS component and the M_DFS component for the combination of the datum feature listed in Tab. 1 can be represented by the corresponding mechanism, and the representation mechanism is a series composition of a fundamental mechanism and an oscillating-bar mechanism. Every combination of the datum feature corresponds to a different fundamental mechanism, such as the crank-guide bar mechanism, the parallelogram mechanism and the variable long crank parallelogram mechanism. The measurement DRF is fixed at the frame of the fundamental mechanism and the design DRF is fixed at the oscillating-bar mechanism. The diagram of the representation mechanism, the calculation schematics and equations of the mechanism parameters are listed in Tab. 2. In Tab. 2, the parameter r is the length of crank of the fundamental mechanism, θ is the rotation angle of the crank of the fundamental mechanism, and λ is the oscillating angle of the oscillating-bar. The parameters D_{D1}, D_{D2}, D_{M1} and D_{M2} denote the diameters of the D_DFS and the M_DFS of the first and second cylindrical datum features; the parameters W_{D1}, W_{D2}, W_{M1} and W_{M2} denote the widths of the D_DFS and the M_DFS of the first and second width datum features. The parameters l and w are the nominal sizes of the width datum feature. All the parameters in calculation equation can be found in the diagram of the representation mechanism or the calculation schematics.

4. The case implementation

A test case is used to demonstrate the implementation of the proposed method, such as shown in Fig. 5, and which is chosen from [5,6,7]. In this example, holes D and E are located at the MMC with respect to the planes A, B and C. Hole G is the related feature, which is located with respect to plane A, hole D and hole E, and both the related feature and datum D and E apply MMC. For the analysis of this particular example, the mechanism model in line 5 of Tab. 2 is used to calculate the ITZ. To demonstrate the benefit of the shift tolerance, the actual matting sizes of both hole D and E are replaced by their least material virtual size (LMVS), and the size of the design tolerance zone of hole G is considered as Ø0.127 mm assuming that hole G be a minimum size Ø4.73 mm.

Table 2 the representation mechanism and parameters

Datum combination	Diagrams of the representation mechanism	Equations of the structural parameter	Kinematic parameters and their calculation schematic	Calculation equations of the kinematic parameters
	Crank guide-bar + oscillating-bar 	$r=\min(r_1,r_2)$ $r_1=\lvert D_{M1}-D_{D1}\rvert/2$ $w=\lvert W_{M2}-W_{D2}\rvert$ $r_2=\frac{w}{\sqrt{w^2+k^2}}l$	 $\theta=0\sim 2\pi$ $\lambda=-\beta\sim\alpha$	$\alpha=\min(\alpha_1,\alpha_2)$ $\alpha_1=\arctan\frac{w-2r\sin\theta}{2l+k-2r\cos\theta}$ $\alpha_2=\arctan\frac{w-2r\sin\theta}{2l-k-2r\cos\theta}$ $\beta=\max(\beta_1,\beta_2)$ $\beta_1=\arctan\frac{-w-2r\sin\theta}{2l+k-2r\cos\theta}$ $\beta_2=\arctan\frac{-w-2r\sin\theta}{2l-k-2r\cos\theta}$
	Variable long crank parallelogram + oscillating-bar 	$r_1=\lvert D_{M1}-D_{D1}\rvert/2$ $w=\lvert W_{M2}-W_{D2}\rvert$ If $2r_1>w$, then $r=\begin{cases} w/(2\cos\theta),\theta\in\delta \\ r_1,\theta\notin\delta \end{cases}$ 	 $\theta=0\sim 2\pi$ $\lambda=-\beta\sim\alpha$	$\alpha=\min(\alpha_1,\alpha_2)$ $\alpha_1=\arctan\frac{h_2}{l}-arc\cos\frac{l+w/2-r\cos\theta}{\sqrt{l^2+h_1^2}}$ $\alpha_2=\arccos\frac{l-w/2-r\cos\theta}{\sqrt{l^2+h_2^2}}-\tan\frac{h_2}{l}$ $\beta=\min(\beta_1,\beta_2)$ $\beta_1=\arctan\frac{h_2}{l}-\arccos\frac{l+w/2-r\cos\theta}{\sqrt{l^2+h_2^2}}$ $\beta_2=\arccos\frac{l-w/2-r\cos\theta}{\sqrt{l^2+h_1^2}}-\tan\frac{h_2}{l}$
	Variable long crank parallelogram + oscillating-bar 	$r_1=\lvert D_{M2}-D_{D2}\rvert/2$ $w=\lvert W_{M1}-W_{D1}\rvert$ If $2r_1>w$, then $r=\begin{cases} w/(2\cos\theta),\theta\in\delta \\ r_1,\theta\notin\delta \end{cases}$ 	 $\theta=0\sim 2\pi$ $\lambda=\pi-\beta\sim\pi+\alpha$	$\alpha=\min(\alpha_1,\alpha_2)$ $\alpha_1=\arctan\frac{h_2}{l}-arc\cos\frac{l+w/2-r\cos\theta}{\sqrt{l^2+h_1^2}}$ $\alpha_2=\arccos\frac{l-w/2-r\cos\theta}{\sqrt{l^2+h_2^2}}-\tan\frac{h_2}{l}$ $\beta=\min(\beta_1,\beta_2)$ $\beta_1=\arctan\frac{h_2}{l}-\arccos\frac{l+w/2-r\cos\theta}{\sqrt{l^2+h_2^2}}$ $\beta_2=\arccos\frac{l-w/2-r\cos\theta}{\sqrt{l^2+h_1^2}}-\tan\frac{h_2}{l}$
	Parallelogram + oscillating-bar 	$r_1=\lvert D_{M1}-D_{D1}\rvert/2$ $r_2=\lvert D_{M2}-D_{D2}\rvert/2$ $r=r_1$	 $\theta=0\sim 2\pi$ $\lambda=-\beta\sim\alpha$	$a=\frac{2l^2+r^2-r_2^2-2lr\cos\theta}{2(l-r\cos\theta)}$ $b=\frac{r\sin\theta}{l-r\cos\theta}$ $\alpha=\arctan\frac{ab+\sqrt{l^2(1+b^2)-a^2}}{a-b\sqrt{l^2(1+b^2)-a^2}}$ $\beta=\arctan\frac{-ab+\sqrt{l^2(1+b^2)-a^2}}{a+b\sqrt{l^2(1+b^2)-a^2}}$
	Parallelogram + oscillating-bar 	$r_1=\lvert D_{M1}-D_{D1}\rvert/2$ $r_2=\lvert D_{M2}-D_{D2}\rvert/2$ $r=r_2$	 $\theta=0\sim 2\pi$ $\lambda=\pi-\beta\sim\pi+\alpha$	$a=\frac{2l^2+r^2-r_2^2-2lr\cos\theta}{2(l-r\cos\theta)}$ $b=\frac{r\sin\theta}{l-r\cos\theta}$ $\alpha=\arctan\frac{-ab+\sqrt{l^2(1+b^2)-a^2}}{a+b\sqrt{l^2(1+b^2)-a^2}}$ $\beta=\arctan\frac{ab+\sqrt{l^2(1+b^2)-a^2}}{a-b\sqrt{l^2(1+b^2)-a^2}}$
	Double slider+ oscillating-bar 	$w_1=\lvert W_{M1}-W_{D1}\rvert$ $w_2=\lvert W_{M2}-W_{D2}\rvert$	 $-w_1/2\le x\le w_1/2$ $-w_2/2\le y\le w_2/2$ $\lambda=-\beta\sim\alpha$	$\alpha=\min(\alpha_1,\alpha_2,\alpha_3)$ $\beta=\min(\beta_1,\beta_2,\beta_3)$ $\alpha_1=\arctan((w_2/2-y)/(l_1+l_2-x))$ $\alpha_2=\arctan((w_1/2+x)/h_2)$ $\alpha_3=\arctan((w_1/2-x)/h_1)$ $\beta_1=\arctan((w_2/2+y)/(l_1+l_2-x))$ $\beta_2=\arctan((w_1/2-x)/h_2)$ $\beta_3=\arctan((w_1/2+x)/h_1)$

The MMVS and LMVS of holes D and E are listed in Tab. 3, and the parameters are l =sqrt((71.18-41.78)^2 + (33.63-29.84)^2) =29.6433mm, r_1 = (18.31-18.163)/2 = 0.0735 mm, and r_2 = (18.591-18.436)/2 = 0.0775 mm.

Table 3 The calculation parameters

Feature	MMVS	LMVS	parameter
D	18.163	18.31	r_1=0.0735
E	18.436	18.591	r_2=0.0775

Fig. 6 shows the enlarged view of the calculation results, which is drawn in the measurement coordinate system x_M-o_M-y_M. The outer outline in solid line is the ITZ, which is a kinematic envelope of the tolerance zone with diameter Ø0.127 mm on the design coordinate system x_D-o_D-y_D. To compare the effectiveness of the oscillating-bar mechanism, the envelope of the design tolerance zone without regard to the oscillating-bar mechanism is drawn; the boundary of the envelope is a circle with diameter Ø0.274 mm, and the circle Ø0.147 mm is the kinematic locus of the center of hole G of the parallelogram mechanism. In fact, the diameter Ø0.274 mm is the sum of the diameter of the design tolerance zone and the minimum value of two diameter difference between the D_DFS and the M_DFS of both hole D and hole E, i.e. 0.274=0.127+2×0.0735. From this example, the minimum value of the shift tolerance is 0.147 mm, and which is larger than the stated tolerance value.

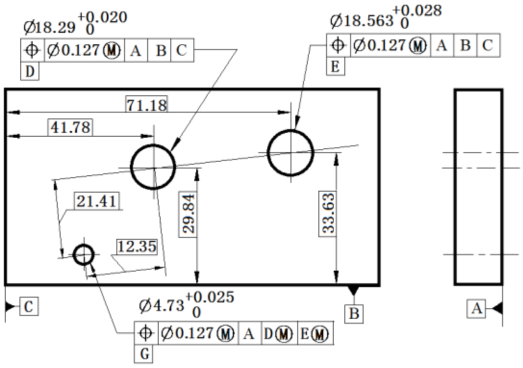


Fig. 5 Example of MMC location

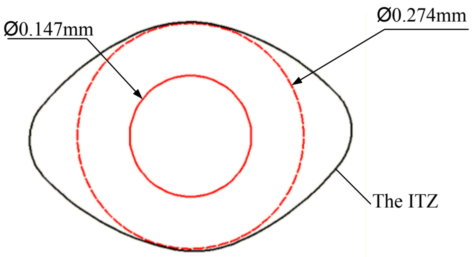


Fig. 6 The ITZ of the case study

5. Conclusions

An analytical method to calculate the ITZ for datum features applying MMC/LMC is developed in this paper, based on the series combination of a fundamental mechanism and an oscillating-bar mechanism. The fundamental mechanism is a crank-guide bar mechanism, or a parallel crank mechanism, a variable long crank parallelogram mechanism and a double slider mechanism etc., based on the combination of two datum features applying MMC/LMC. The mechanism model simulates the movements of the functional gage with respect to the actual datum features, which capture the meaning of the shift tolerance and is easy understood by the engineering designers. The method presented is generally applicable for the calculation of ITZ when two datum features applying MMC/LMC, which facilitates its decoding by analysis, tolerancing and metrology software.

The ITZ is calculated based on the associated geometries of a related feature and datum feature, and the associated geometries are from the numerical calculation of extracted features. To use the calculation method of the ITZ, the numerical calculation must be conducted at first.

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References

[1] ASME. Mathematical Definition of Dimensioning and Tolerancing Principles - Engineering Drawing and Related Documentation Practices. ASME Y14.5.1M-1994.
[2] ISO 2692:2014. Geometrical product specifications (GPS) - Geometrical tolerancing - Maximum material requirement (MMR), least material requirement (LMR) and reciprocity requirement (RPR). 2014.
[3] Jayaraman R, Srinivasan V. Geometric tolerancing: 1. virtual boundary requirements. IBM Journal of Research and Development 1989; 33(2): 90-104.
[4] Anselmetti B. Tolerancing method for function and manufacturing. Proceedings of ILCE 1995.
[5] Robinson DM. Geometric tolerancmg for assembly with maximum material parts. Proceedings of the 5th CIRP CAT 1997; 242-53.
[6] Bennis F, Pino L, Fortin C. Geometric tolerance transfer for manufacturing by an algebraic method. Proceedings of the 2th IDMME 1999; 373-80.
[7] Pino L, Bennis F, Fortin C. The use of a kinematic model to analyze positional tolerances in assemblies. Proceeding of the 1999 IEEE Conference on Robotics and Automation 1999; (2):1418-23.
[8] Pino L, Bennis F, Fortin C. Calculation of virtual and resultant part for variational assembly analysis. Proceedings of the 3th IDMME 2003; 89-92.
[9] Pairel E, Hernandez P, Giordano M. Virtual Gauges Representation for Geometrical Tolerances in CAD-CAM Systems. Proceedings of the 9th CIRP CAT 2007; 3-12.
[10] Dantan JY. Functional and product specification by Gauge with Internal Mobilities (GIM). Proceedings of the 6th CIRP CAT1999; 37-46.
[11] Dantan JY, Ballu A. Assembly specification by Gauge with Internal Mobilities (GIM) -A specification semantics deduced from tolerance synthesis. Journal of Manufacturing Systems 2002; 21(3):218-235.
[12] Shen Z, Shah JJ, Davidson JK. Simulation-based tolerance and assemblability analysis of assemblies with multiple pin/hole floating mating conditions. Proceedings of ASME IDETC/CIE 2005; 429-40.
[13] Shen Z, Shah JJ, Davidson JK. A complete variation algorithm for slot and tab features for 3D simulation-based tolerance analysis. Proceedings of ASME IDETC/CIE 2005; 587-97.
[14] Robin C, Anselmetti B. Functional tolerancing: Virtual material condition on complex junctions. Computers in Industry 2012; 63:210-21.
[15] Anselmetti B, Pierre L. Complementary writing of maximum and least material requirements with an extension to complex surfaces. Procedia CIRP 2016; 43:220-5.
[16] ASME. Dimensioning and Tolerancing - Engineering Drawing and Related Documentation Practices. ASME Y14.5M-2009.