A Comparison between the Four- and Five-Point Double-Toggle Mould Clamping Mechanisms

Wen-Yi Lin

Assistant Professor, Department of Mechanical Engineering, De Lin Institute of Technology

Abstract

The comparison between the conventional five- and four-point toggle mechanisms available in the literature may be insufficiently objective and fair, because the variations of the offset, the crosshead height, the projections of the moving and tailstock platens and the necessary thrust applied to the crosshead are not considered simultaneously. A comparison of the four-point type is performed with the conventional and Fanuc five-point types for the larger stroke ratio and thrust saving cases, on condition that the overall horizontal length cannot exceed a given one. A quantitative exhaustive search algorithm is presented to find the optimal dimension after dimensional synthesis. An easy method, the 2-D movable sketch of SolidWorks, is used to check there is no circuit defect for the optimal dimensions.

Keywords: injection-moulding machine, toggle clamping mechanism

四點式與五點式雙肘節合模機構之比較分析

林文一

德霖技術學院機械工程系副教授

摘　要

工業界或文獻上對射出機或壓鑄機的傳統四點式與傳統五點式雙肘節鎖模機構的一般認識為：在相同輸出行程下，傳統四點式所需的輸入行程大於傳統五點式，但傳統四點式比傳統五點式較省力。由於沒有同時考量到偏置、十字頭高度與模板的法蘭長度的變化以及將十字頭所需推力同時納入考量，因此在這些比較中，不夠深入與客觀。本文在機構總長不能超過給定值下，針對給定的輸入與輸出行程不變的條件下，找出四點式與傳統五點式以及 Fanuc 專利五點式具有最小推力的尺寸，並比較此三種機構的省力性能；此外又針對在十字頭推力及機構總長不能超過給定值下，找出此三種機構具有最大行程比的尺寸，比較了此三種機構的行程比的性能。本文採用 quantitative exhaustive search algorithm 找出尺寸合成後的最佳尺寸，並藉由 SolidWorks 2D 運動骨架圖檢驗得到的最佳尺寸沒有迴路缺陷。

關鍵字：射出成型機、肘節合模機構

Ⅰ. Introduction

The conventional five-point double-toggle mould clamping mechanism is extensively used for small- to middle-sized injection-moulding machines **[1]**. Fanuc Japan Company Limited developed a new five-point double-toggle mould clamping mechanism **[2]**. The conventional four-point double-toggle mould clamping mechanism is shown in Fig 1, where the upper portion above the centre-line (CL) illustrates the mould closing state, while the lower portion below the centre-line illustrates the initial state. Let numerals 1 to 8 shown in Fig. 1 denote moving-platen side links (links 1), tailstock-platen side links (links 2), crosshead links (links 3), crosshead (4), tailstock platen (5), moving platen (6), tie bars (7) and stationary platen (8) respectively in this work. Points A to E denote centres of pin joints.

In reference **[3]**, an elastostatic model having the mechanical advantage along with Coulomb friction for the conventional and Fanuc five-point types is presented to determine the necessary thrust applied to the crosshead. Besides, with keeping the value of (L_1+L_2) (L_1 and L_2 are the distances between points A and B and points B and C respectively) and the heights of the three sliders (the tailstock and moving platens and the crosshead) and the final orientation of the crosshead link invariable, a parameter study is proposed to perform dimensional synthesis for the conventional and Fanuc five-point toggle linkages and to find the optimal dimensions in the programming discrete domain. Such an approach to find optimal dimensions may be classified as the exhaustive search algorithm. It is concluded in reference **[3]** that the ratio of L_1 to L_2, whether for the conventional type or for the Fanuc type, should be as small as possible for thrust saving on condition that the geometry can be satisfied and the transmission characteristic can be accepted. Furthermore, the angle of swing of links 2 becomes smaller for the same output stroke, so that the horizontal length of the projection (flange) of the tailstock platen may be reduced. If horizontal space is not enough for the ejector unit, the reference **[3]** adopts lengthening the projection of the moving platen. The variation of the overall horizontal length of the mechanism may be acceptable. Reference **[4]** attempts to enhance the performances of the stroke ratio and/or thrust saving of the conventional and Fanuc five-point double-toggle mould clamping mechanisms, at the cost of increasing the offset and varying the crosshead height and loosening the initial transmission performance from 3.1 per cent of the original maximum thrust of reference **[3]** to 6.0 per cent, on condition that the overall horizontal length cannot exceed the one of the original design. The results without loosening the initial transmission performance are also presented. An objective and fair comparison is showed between the conventional and Fanuc five-point types. Some features of the profiles of the toggle linkages with the larger stroke ratio and/or thrust saving are illustrated.

The conventional four-point type has now largely given way to the five-point type due to the following recognition **[5]**. As pointed out in references **[1, 6]**, for the same output stroke, the input stroke of the conventional four-point type is greater than the one of the conventional five-point type and therefore the overall horizontal length of the former is greater than that of the latter. It is shown in reference **[7]** that an output stroke of the conventional four-point type is greater than that of the conventional five-point type, but the stroke ratio of the former is less than that of the latter. In other words, for a given horizontal length of the mechanism, the conventional five-point type can be made to fold back within a shorter distance (input stroke) than the conventional four-point type and the former provides a longer output stroke than the latter [2, 5]. However, previous comparisons between the conventional five- and four-point types available in the literature may be insufficiently objective or unfair because the variations of the offset, the crosshead height, the projections of the moving and tailstock platens and the necessary thrust applied to the crosshead are not considered simultaneously.

The aim of this work is to perform a comparison of the four-point type with the conventional and Fanuc five-point types for the larger stroke ratio and thrust saving cases, on condition that the overall horizontal length cannot exceed the one of the original design. An elastostatic model and then a parameter study for the four-point type are presented in this work, much as the elastostatic model and the parameter study for the five-point type were presented in references [3, 4]. The original design of the conventional five-point type in reference [3] serves as the base of the comparison. The present results for the four-point type with and without loosening the original initial transmission performance are presented and compared with these for the conventional and Fanuc five-point types in reference [4]. For the comparison of thrust saving, the input and output strokes are the same as these of the original design in reference [3]. On the other hand, for the comparison of the larger stroke ratio, the necessary maximum thrust should not exceed that of the original design.

Ⅱ. Elastostatic model for the four-point type

In the course of the real-mould clamping, in order to develop the clamping force, the toggle clamping mechanism must overcome the friction forces in pin joints (hinge friction) and slider connections, and the total deformation force of the tie bars. Here, the slider connections comprise the crosshead and the guide rods, and the tailstock platen and the (precision) ground steel bands. The following assumptions are made: 1. The axial deformation displacements of links and tie bars are small, and the flexural deformations of links and tie bars are negligible. 2. The inertia forces, the weights and the friction forces in the slider connections can be neglected when compared to the total deformation force of the tie bars and the thrust applied to the crosshead. 3. The deformation effects of the mould and the mould platens are negligible. 4. Coulomb friction is valid for the friction in pin joints. In this paper, the symbol $\tilde{(\ )}$ denotes that the quantity in parentheses is in the state of the final position of the mould closing operation.

Figure 2 depicts a skeleton drawing for the lower half of the four-point double-toggle clamping mechanism during the real-mould clamping operation. The dashed lines denote the position when the moving mould is just in contact with the stationary mould. From Fig. 2, the geometry of the final position of the real-mould clamping operation and the assumption of small deformation give

$$L_2\sin\alpha_c - L_1\sin(\pi-\beta_c) = (L_2-\delta_{BC})\sin\alpha - (L_1-\delta_{AB})\sin(\pi-\beta) = d_A \tag{1}$$

$$L_2\sin\alpha_c + L_3\sin(\pi-\phi_c) = (L_2-\delta_{BC})\sin\alpha + (L_3-\delta_{BE})\sin(\pi-\phi) = d_E \tag{2}$$

$$\tilde{\beta} = \tilde{\alpha} + \pi \tag{3}$$

$$\begin{aligned} U_C &= (L_1-\delta_{AB})\cos(\pi-\beta) - L_1\cos(\pi-\beta_c) + (L_2-\delta_{BC})\cos\alpha - L_2\cos\alpha_c \\ &= \overline{U}_C - \delta_{AB}\cos(\pi-\beta) - \delta_{BC}\cos\alpha \end{aligned} \tag{4}$$

$$\overline{U}_C = L_1[\cos(\pi-\beta) - \cos(\pi-\beta_c)] + L_2(\cos\alpha - \cos\alpha_c) \tag{5}$$

where L_i ($i = 1 - 3$) are the distances between points A and B, B and C and B and E respectively at the undeformed state of the mechanism; U_C is the elongation of the tie bars; and δ_{AB}, δ_{BC} and δ_{BE} are the axial shortened lengths of links 1, 2 and 3 respectively. Due to assumption of small deformation, δ_{AB}, δ_{BC} and δ_{BE} in equations (1) and (2) are dropped in this study.

During the real-mould clamping process, links 1, 2 and 3 are subjected to forces F_1, F_2 and F_3 respectively, as shown in Fig. 3. The circles shown in Fig. 3 are called friction circles **[8, 9]**. For the sake of clarity, the friction circles in Fig. 3 have been greatly exaggerated in magnitude. From Figs 1 and 3, the free-body diagrams for each member and joint can be easily drawn (not shown) and the equations of equilibrium required for the real-mould clamping operation are given as follows:

$$F_{cl} = F_1 \cos(\pi - \beta + \beta_\mu) \tag{6}$$

$$F_o = F_3 \cos(\pi - \phi - \phi_\mu) \tag{7}$$

$$F_c = F_2 \cos(\alpha + \alpha_\mu) \tag{8}$$

$$F_1 \sin(\pi - \beta + \beta_\mu) + F_2 \sin(\alpha + \alpha_\mu) - F_3 \sin(\pi - \phi - \phi_\mu) = 0 \tag{9}$$

$$F_1 \cos(\pi - \beta + \beta_\mu) - F_2 \cos(\alpha + \alpha_\mu) - F_3 \cos(\pi - \phi - \phi_\mu) = 0 \tag{10}$$

$$\alpha_\mu = \sin^{-1}(\frac{2\rho_B}{L_2}), \quad \beta_\mu = \sin^{-1}(\frac{2\rho_B}{L_1}), \quad \phi_\mu = \sin^{-1}(\frac{\rho_B + \rho_E}{L_3}) \tag{11}$$

$$\rho_B = \frac{\mu}{\sqrt{1+\mu^2}} r_B, \quad \rho_E = \frac{\mu}{\sqrt{1+\mu^2}} r_E \tag{12}$$

where F_{cl} is the clamping force, F_o is the thrust applied to the crosshead and F_c is the total tension force of the tie bars; μ is the friction coefficient in pin joints; r_B and r_E are the radii of pins at joints B and E respectively, and ρ_B and ρ_E are the corresponding friction radii respectively. The radii for pins A, B and C are the same.

The compressive axial forces in a single link 1 and 2 can be expressed as

$$F_{AB} = \frac{F_{cl} \cos\beta_\mu}{n_1 \cos(\pi - \beta + \beta_\mu)}, \quad F_{BC} = \frac{F_c \cos\alpha_\mu}{n_2 \cos(\alpha + \alpha_\mu)} \tag{13}$$

where n_1 and n_2 are the number of links 1 and 2 respectively.

The axial shortened lengths of links 1 and 2 can be expressed as

$$\delta_{AB} = \frac{F_{AB} L_1}{A_1 E_1}, \quad \delta_{BC} = \frac{F_{BC} L_2}{A_2 E_2} \tag{14}$$

where A_i and E_i ($i = 1, 2$) are the cross-sectional area and Young's modulus of links 1 and 2 respectively. The elongation of the tie bars can be expressed by

$$U_C = \frac{F_c L_c}{n_c A_c E_c} \tag{15}$$

where L_c is the length of the tie bar, n_c is the number of the tie bar, and A_c and E_c are the cross-sectional area and Young's modulus of the tie bar respectively.

Substituting equations (14) – (15) into equation (4) gives

$$\frac{F_c L_c}{n_c A_c E_c} + \frac{F_{AB} L_1}{A_1 E_1} \cos(\pi - \beta) + \frac{F_{BC} L_2}{A_2 E_2} \cos\alpha = \overline{U}_C \tag{16}$$

Using equations (6), (7), (9) and (10), the mechanical advantage M_a of the four-point toggle clamping mechanism can be obtained as

$$M_a = \frac{F_{cl}}{F_o} = \frac{\tan(\alpha + \alpha_\mu) + \tan(\pi - \phi - \phi_\mu)}{\tan(\alpha + \alpha_\mu) + \tan(\pi - \beta + \beta_\mu)} \tag{17}$$

From equations (6), (7), (8) and (10),

$$F_c + F_o = F_{cl} \tag{18}$$

After substituting equation (13) into equation (16), the relationship of the total deformation force of the tie bars F_c, the clamping force F_{cl} and the necessary thrust applied to the crosshead F_o can be obtained from equations (16), (17) and (18). For a specified final total deformation force of the tie bars, the final clamping force and the thrust applied to the crosshead can be obtained from equations (17) and (18). Then, the final deformations for links 1, 2 and tie bars can be calculated from equations (13) – (15). Then, using equations (1), (2), (4) and (5), the values of α_c, β_c and ϕ_c required at the beginning of the real-mould clamping operation can be determined.

Ⅲ. Interferences and overall length of the mechanism

The crosshead is thrust by means of a drive mechanism composed of a servo electric motor and a ball nut-screw. The motor can actuate the screw shaft of the ball nut-screw to rotate for driving the crosshead locked on the nut of the ball nut-screw to move in the centre-line of the machine. For the four-point type, to prevent the interference between the toggle linkages and the screw shaft of the ball nut-screw, the following condition with a minimum clearance 5 mm should be satisfied.

$$\begin{array}{ll} L_2 \sin\alpha_0 + R_B \le d_0 - R_{bs} - 5 & \text{if} \quad \alpha_0 < \dfrac{\pi}{2} \\ L_2 + R_B \le d_0 - R_{bs} - 5 & \text{if} \quad \alpha_0 \ge \dfrac{\pi}{2} \end{array} \tag{19}$$

where R_B is the radius of joint B and R_{bs} denotes the major radius of the screw shaft.

In addition, to prevent the interference between joint B and the crosshead in the initial position, the following condition with a minimum clearance 5 mm should be satisfied.

$$L_3 \cos(\phi_0 - \pi) \ge R_B + 5 \tag{20}$$

The following geometry conditions may be satisfied for the four-point type.

$$L_1 \ge 2R_B,\quad L_2 \ge 2R_B \quad \text{and} \quad L_3 \ge R_B + R_E \tag{21}$$

where R_E is the radius of joint E.

The detailed expression for the variation of the overall horizontal length of the mechanism ΔT_h shown in reference **[4]** is employed. Moreover, $\Delta T_h \le 0$ is demanded.

Ⅳ. Initial transmission performance

Most of the injection-moulding machines have three-step to five-step adjustment for the mould closing speed. The initial thrust applied to the crosshead is about 3.1 per cent of the necessary maximum thrust for the original mechanism with constant acceleration 120 $\mathrm{mm/s^2}$ in the crosshead to 0.5 s for the first stage of the mould closing process. To avoid unsound motion characteristic and consider the initial transmission characteristic for a fair comparison and a further enhancement on thrust saving, the following conditions are demanded individually

$$a_A^0 > 0 \text{ and } 0 < \frac{F_o^0}{F_{o,\max}^*} \le 3.1\%$$

$$a_A^0 > 0 \text{ and } 0 < \frac{F_o^0}{F_{o,\max}^*} \le 6\% \tag{22}$$

where a_A^0 denotes the initial acceleration of the moving platen; F_o^0 denotes the initial thrust applied to the crosshead; and $F_{o,\max}^*$ denotes the necessary maximum thrust for the original conventional five-point mechanism. The superscript * is used only for the original mechanism shown in reference **[3]**. The mass of the moving platen with the ejector unit and the moving mould is 220 kg. The static friction coefficients at the slider connection for the moving platen and pin joints are both 0.125. An easy method, the 2-D movable sketch (i.e. movable stick diagram) of SolidWorks, is used to check there is no circuit defect for the optimal dimensions from the discrete exhaustive search algorithm proposed in this study.

V. Quantitative exhaustive search algorithm

A conventional five-point double-toggle mechanism used in an injection-moulding machine serves as the base of comparisons. The geometric and material properties of the original mechanism are shown in Table 1. For the four-point type, $n_1 = 4$ and $A_1 = 1.5A_1^*$. Without changing the specifications of the mould clamping force and the ejector unit, the original allowable mould thickness and the stationary mould platen for the base of comparisons, let A_2, A_c, d_f, e_2, E_1, E_2, E_c, $\tilde{h}_{bsE}$, n_2, n_c, r_B, r_C, r_D $(=r_E$ for the four-point type), R_{bs}, R_B, R_D $(=R_E$ for the four-point type), R_o, T_m, T_s (see Fig. 1) and $L_1 + L_2$ be invariable in the parametric study. Moreover, the friction coefficient in the pin joints during the real-mould clamping process is assumed to be 0.1 and the final total deformation force of the tie bars $\tilde{F}_c = 539$ kN is considered.

In this study, the values of d_A, d_E, L_1, L_2, $\tilde{S}_A$ (output stroke) and $\tilde{S}_E$ (input stroke) are prescribed in this study. Thus, the values of $\tilde{\alpha}$ and $\tilde{\beta}$ can be obtained. The value of α_0 can be solved easily using the bisection method to the following equation of the output stroke derived very straightforward from the kinematic diagram for the initial and final positions.

$$\tilde{S}_A = \sqrt{(L_1 + L_2)^2 - d_A^2} - \sqrt{L_1^2 - (L_2 \sin\alpha_0 - d_A)^2} - L_2 \cos\alpha_0 \tag{23}$$

Obviously, the value of β_0 can also be obtained. In this study, let $\tilde{\phi} \ge 91^\circ$. The value of $\tilde{\phi}$ can be solved easily using the bisection method to the following equation of the input stroke derived quite straightforward from the kinematic diagram for the initial and final positions.

$$\tilde{S}_E = \sqrt{[\frac{d_E - L_2 \sin\tilde{\alpha}}{\sin(\pi - \tilde{\phi})}]^2 - (d_E - L_2 \sin\alpha_0)^2} - (d_E - L_2 \sin\tilde{\alpha})\cot(\pi - \tilde{\phi}) + L_2(\cos\tilde{\alpha} - \cos\alpha_0) \qquad (24)$$

Obviously, the value of L_3 and then the value of ϕ_0 can be obtained

Let $\tilde{S}_A = \tilde{S}_A^*$ and $\tilde{S}_E = \tilde{S}_E^*$ (i.e. the stroke ratio $S_r = S_r^* = 0.84$) for the thrust saving case. For the larger stroke ratio case, let $\tilde{S}_A = 1.25\tilde{S}_A^*$ and $\tilde{S}_E = 0.85\tilde{S}_E^*$ ($S_r = 1.47 S_r^* = 1.23$) for the four-point type. Also let d_0 be 235 – 300 mm with an increment 5 mm and the corresponding offset d_A be $d_0 - (d_0^* - d_A^*)$ and then let d_E vary from ($d_0 - 95$) to 80 mm with a decrement 5 mm. For above cases, let L_1/L_2 be 1.7 to 0.6 with a decrement 0.01 to find the profiles of the toggle linkages with the minimum value of $F_{o,\max}$ denoted by $F_{o,\max}^g$ in the larger stroke ratio and thrust saving cases.

Ⅵ. Results and discussion

The geometric parameters and the initial thrust corresponding to $F_{o,\max}^g$ for the four- and five-point types with $F_o^0 / F_{o,\max}^* \le 3.1\%$ and $F_o^0 / F_{o,\max}^* \le 6\%$ for the thrust saving case and the larger stroke ratio case are shown in Tables 2 and 3 respectively. The results of the conventional and Fanuc five-point types are from reference **[4]**.

A.The thrust saving case

For $F_o^0 / F_{o,\max}^* \le 3.1\%$, the thrust $F_{o,\max}^g$ for the four-point type is about 12 per cent of $F_{o,\max}^*$ smaller than the one for the conventional five-point type; moreover, the height of the linkage above CL for the latter increases by 30 mm as compared with the former. On the other hand, the thrust $F_{o,\max}^g$ for the four-point type is about 7 per cent of $F_{o,\max}^*$ greater than the one for the Fanuc five-point type; moreover, the height of the linkage above CL for the latter increases by 64 mm as compared with the former.

For $F_o^0 / F_{o,\max}^* \le 6\%$, the thrust $F_{o,\max}^g$ for the four-point type is about 6 per cent of $F_{o,\max}^*$ smaller than the one for the conventional five-point type; moreover, the height of the linkage above CL for the latter increases by 30 mm as compared with the former. On the other hand, the thrust $F_{o,\max}^g$ for the four-point type is about 4 per cent of $F_{o,\max}^*$ greater than the one for the Fanuc five-point type;

moreover, the height of the linkage above CL for the latter increases by 58 mm as compared with the former. The kinematic diagram in the open and closed positions for the conventional five- and four-point types with $F_o^0 / F_{o,\max}^* \le 6\%$ is shown in Fig. 4a. The horizontal length of the projection of the tailstock platen for the four-point type decreases by 60.0 mm as compared to the conventional five-point type. On the other hand, the horizontal length of the projection of the moving platen for the former increases by 57.7 mm as compared to the latter.

For showing the effect of the size of discrete section regarding L_1 / L_2 (from 1.7 to 0.6), decrements with 0.01, 0.005, 0.002 and 0.001 are performed. The corresponding results of $F_{o,\max}^g / F_{o,\max}^*$ for the four-point type are 67.95, 67.83, 67.86 and 67.83 per cent respectively appearing in the same offset and crosshead height.

B. The larger stroke ratio case

For $F_o^0 / F_{o,\max}^* \le 6\%$, the stroke ratio for the conventional five-point type is about 7 per cent smaller than the one for the four-point type, while the thrust $F_{o,\max}^g$ for the former is about 0.7 per cent of $F_{o,\max}^*$ smaller than the one for the latter. Moreover, the initial transmission performance for the latter is slightly superior to the former. On the other hand, the stroke ratio for the Fanuc five-point type is about 11 per cent greater than the one for the four-point type, and the thrust $F_{o,\max}^g$ for the former is about 3 per cent of $F_{o,\max}^*$ smaller than the one for the latter. Moreover, the height of the linkage above CL for the former increases by about 21 mm as compared with the latter. The kinematic diagram in the open and closed positions for the conventional five- and four-point types with $F_o^0 / F_{o,\max}^* \le 6\%$ is shown in Fig. 4b. The horizontal length of the projection of the tailstock platen for the four-point type decreases by 49.4 mm as compared to the conventional five-point type. On the other hand, the horizontal length of the projection of the moving platen for the former increases by 51.5 mm as compared to the latter.

The values of $F_{o,\max}^g / F_{o,\max}^*$ for the four-point type are 94.89, 92.87, 91.77 and 91.77 per cent appearing in the same offset and crosshead height for L_1 / L_2 from 1.7 to 0.6 with decrements 0.01, 0.005, 0.002 and 0.001 respectively. Notice that crosshead links, links 1 and links 2 should be situated from inner to outer position, respectively (so crosshead links situated in the middle position from the top view), in order to prevent the interference between the crosshead and links 2.

Ⅶ. Conclusions

By the variations of the offset and the crosshead height, and adjustments of the horizontal lengths of the projections of the tailstock and moving platens, and the parameter study, the present comparison reverses the previous recognition that the conventional five-point type can be made to fold back within a shorter input stroke than the conventional four-point type and the former can provide a longer output stroke than the latter for a given horizontal length of the mechanism. Both of the stroke ratio and the initial transmission performance for the four-point type are slightly superior to the conventional five-point type, with the almost same necessary maximum thrust. On the other hand, both of the stroke ratio and thrust saving for the Fanuc five-point type are superior to the four-point type at the cost of the linkage height and the initial transmission performance.

For the same input and output strokes, both of the thrust saving and the linkage height for the four-point type are superior to the conventional five-point type. On the other hand, for the same input and output strokes, the performance of thrust saving for the Fanuc five-point type is superior to the four-point type at the cost of the linkage height.

There are several features of the four-point double-toggle linkages with $F_o^0 \big/ F_{o,\max}^* \le 6\%$ stated in the following. 1. The ratio of L_1 to L_2 are smaller than one for the thrust saving case and close to one for the larger stroke ratio case. 2. The decrement in the horizontal length of the projection of the tailstock platen for the four-point type is greater than the one for the conventional five-point type. On the other hand, the increment in the horizontal length of the projection of the moving platen for the former is greater than the one for the latter. 3. For the larger stroke ratio case, the crosshead height should increase with increasing the stroke ratio.

Nomenclature

a_A	acceleration of the moving platen
A_1, A_2, A_c	cross-sectional areas of link 1, link 2 and the tie bar
d_0	vertical distance between the centre-line of the machine and point C
d_A	vertical distance between points C and A
d_{ch}	semi-height of the crosshead
d_E	vertical distance between points C and E
d_f	vertical distance between the centre-line of the machine and the upper surface of the frame
d_v	height of the linkage above the centre-line (CL) of the machine
e_1	horizontal distance between point A and the left end of the ejector unit
e_2	vertical distance between the centre-line of the machine and the bottom end of the ejector unit
E_1, E_2, E_c	Young's Modulus of links 1, links 2 and the tie bar
F_{AB}, F_{BC}	axial forces in a single link 1 and a single link 2
F_{cl}	clamping force
F_i $(i = 1 - 3)$	forces exerted on all the member i
F_o	thrust applied to the crosshead
$F_{o,\max}$	maximum value of the thrust F_o during the mould clamping process
$F_{o,\max}^g$	minimum value of $F_{o,\max}$ in the thrust saving or larger stroke ratio case

$\tilde{h}_{bsE}$	distance between point E and the right end of the ball nut-screw in the final position
h_{CE}^{0}	horizontal distance between points E and C in the initial position
L_3	distance between points B and E for the four-point type or between points D and E for the five-point type
L_c	length of the tie bar under tension
L_i $(i = 1, 2, 4)$	distances between points A and B, B and C and C and D (point D and L_4 for the five-point type shown in the work of Lin et al.)
M_a	mechanical advantage
n_1, n_2, n_c	number of links 1, 2 and the tie bar
r_B, r_C, r_D, r_E	radii of pins at joints B, C , D and E (pin D for the five-point type)
R_B	semi-width of links 1 or radius of joint B shown in Fig. 1
R_D	semi-width of links 3 for the five-point type
R_E	radius of joint E for the four-point type shown in Fig. 1
R_o	dimension shown in the work of Lin et al. (2006)
S_A, S_E	displacements of the moving platen and the crosshead
S_r	stroke ratio defined as the ratio of the output stroke to the input stroke
T_h, T_m, T_s	dimensions defined in Fig. 1
U_C	elongation of the tie bars
$\overline{U}_C$	parameter defined in equation (5)
α, β, ϕ	angles defined in Fig. 2
α_0, β_0, ϕ_0	angles α, β and ϕ in the initial position
α_c, β_c, ϕ_c	angles α, β and ϕ in the position when the moving mold is just in contact with the stationary mold
γ_C	angle defined in the five-point type
Δd_v	$d_v - d_v^*$
$-\Delta h$	variation of the horizontal length of the projection of the moving platen
Δh_{CE}^{0}	variation of the horizontal length of the projection of the tailstock platen
ΔT_h	$T_h - T_h^*$
$(\ )^0$	quantity in the state of the initial position of the mold closing process
$\tilde{(\ )}$	quantity in the state of the final position of the mould closing process
$(\ )^*$	quantity only for the original mechanism

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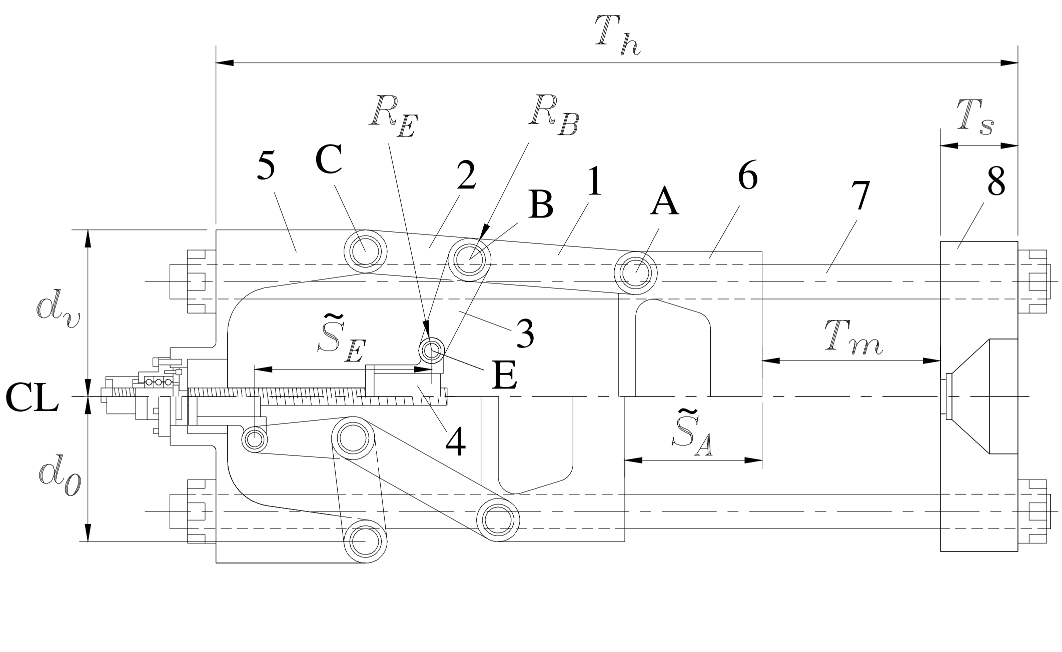

Fig. 1 Four-point double-toggle mould clamping mechanism

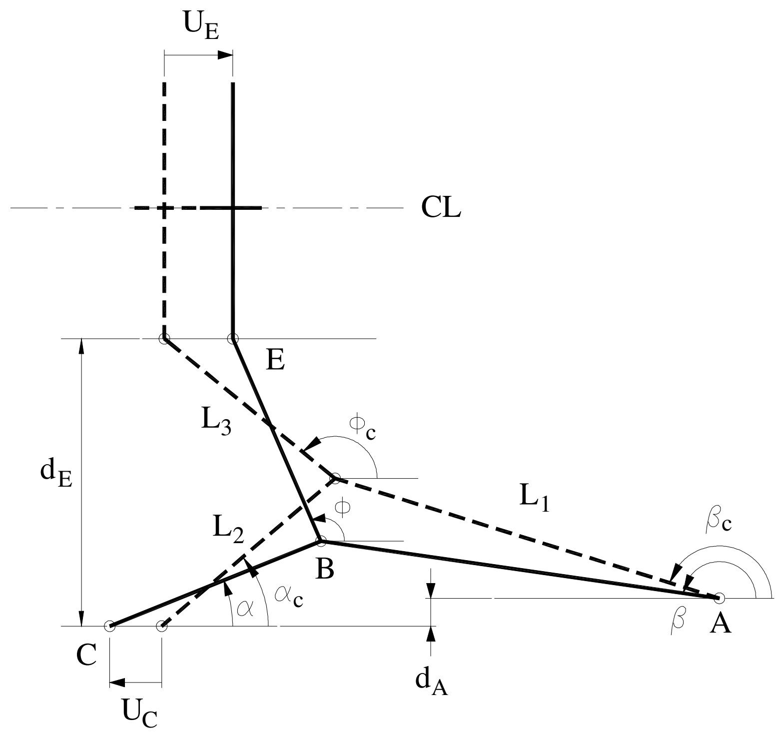

Fig. 2　Geometry of the toggle linkage during the real-mould clamping

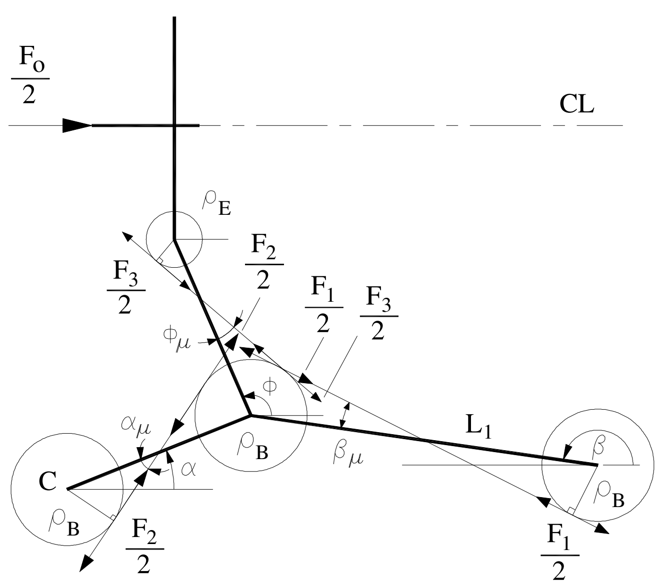

Fig. 3　Toggle linkage subjected to loading during the real-mould clamping process

Table 1 Geometric and material properties of the original conventional five-point mechanism

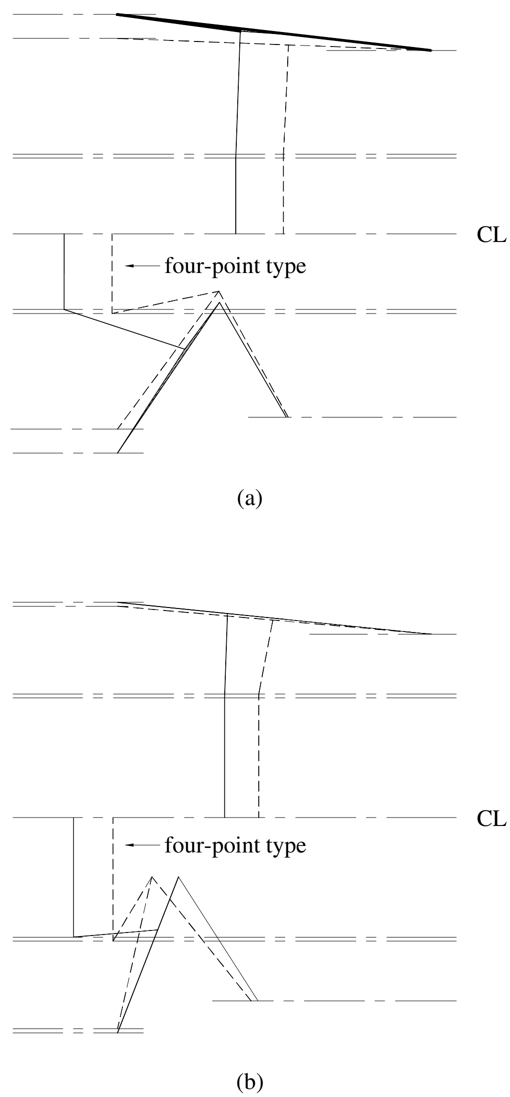

Fig. 4 Kinematic stick diagrams for the conventional five- and four-point types in the open and closed positions: (a) the thrust saving case and (b) the larger stroke ratio case

Parameter	Value	Parameter	Value	Parameter	Value	Parameter	Value
A_1	2184	d_v^*	277	L_2^*	164	R_{bs}	25
A_2	3276	e_1^*	47	L_3^*	70.04	R_B	42
A_c	2827	e_2	81	L_4^*	133.17	R_D	29
d_0^*	235	E_1，E_2	172	L_c^*	1250	R_o	60
d_A^*	5	E_c	204	n_1	6	$\tilde{S}_A^*$	180.67
d_{ch}^*	129	$\tilde{h}_{bsE}$	35	n_2，n_c	4	$\tilde{S}_E^*$	215.23
d_E^*	135	h_{CE}^{0*}	101.44	r_B，r_C	22.5	T_h^*	1250
d_f	350	L_1^*	231	r_D	15	γ_C^*	28.49°

Units: A_1, A_2 and A_c in mm^2, E_1, E_2 and E_c in GPa and the length and distance in mm

Table 2 Geometric parameters corresponding to $F_{o,\max}^{g}$ for the thrust saving case

Conventional four-point type with $\tilde{S}_A = \tilde{S}_A^*$ and $\tilde{S}_E = \tilde{S}_E^*$ ($S_r = S_r^* = 0.84$)											
d_0	d_A	d_E	$\frac{L_1}{L_2}$		$\tilde{\phi}$ (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 / F_{o,\max}^{*} \le 3.1\%$											
235	5	115	1.31		91.93	112.90	76.30	3.1	-13.5	0	149
$F_o^0 / F_{o,\max}^{*} \le 6\%$											
245	15	145	0.84		92.62	136.99	67.95	5.9	-13.5	10	129
Conventional five-point type with $\tilde{S}_A = \tilde{S}_A^*$ and $\tilde{S}_E = \tilde{S}_E^*$ ($S_r = S_r^* = 0.84$) [**4**]											
d_0	d_A	d_E	$\frac{L_1}{L_2}$	$\frac{L_4}{L_2}$	γ_C (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 / F_{o,\max}^{*} \le 3.1\%$											
265	35	155	0.97	0.99	20.52	69.27	88.25	3.1	-13.5	30	139
$F_o^0 / F_{o,\max}^{*} \le 6\%$											
275	45	180	0.73	0.68	1.19	159.20	74.13	5.9	-13.5	40	124
Fanuc five-point type with $\tilde{S}_A = \tilde{S}_A^*$ and $\tilde{S}_E = \tilde{S}_E^*$ ($S_r = S_r^* = 0.84$) [**4**]											
d_0	d_A	d_E	$\frac{L_1}{L_2}$	$\frac{L_4}{L_2}$	γ_C (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 / F_{o,\max}^{*} \le 3.1\%$											
235	5	110	1.54	1.0	30.45	187.22	69.58	3.1	-13.5	64.1	154
$F_o^0 / F_{o,\max}^{*} \le 6\%$											
245	15	135	1.02	0.90	25.92	205.98	64.23	5.1	-13.5	67.9	139

$L_1 + L_2 = 395$ mm, $\tilde{F}_c = 539$ kN, $F_{o,\max}^{*} = 16$ kN, $T_h^* = 1250$ mm, $d_v^* = 277$ mm, $\tilde{\phi} = 92$ for the five-point type

Table 3　Geometric parameters corresponding to $F_{o,\max}^{g}$ for the larger stroke ratio case

Conventional four-point type with $\tilde{S}_A = 1.25\tilde{S}_A^*$ and $\tilde{S}_E = 0.85\tilde{S}_E^*$ ($S_r = 1.47 S_r^* = 1.23$)

d_0	d_A	d_E	$\frac{L_1}{L_2}$	$\tilde{\phi}$ (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 \big/ F_{o,\max}^{*} \le 3.1\%$										
The result has the necessary maximum thrust exceeding the original one by 5.17%.										
$F_o^0 \big/ F_{o,\max}^{*} \le 6\%$										
265	35	110	1.02	100.72	94.32	94.89	4.1	-0.6	30	184

Conventional five-point type with $\tilde{S}_A = 1.2\tilde{S}_A^*$ and $\tilde{S}_E = 0.88\tilde{S}_E^*$ ($S_r = 1.36 S_r^* = 1.14$) [**4**]

d_0	d_A	d_E	$\frac{L_1}{L_2}$	$\frac{L_4}{L_2}$	γ_C (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 \big/ F_{o,\max}^{*} \le 3.1\%$											
No solution can satisfy the interference condition or the initial transmission performance, etc.											
$F_o^0 \big/ F_{o,\max}^{*} \le 6\%$											
270	40	120	0.88	0.66	0.05	105.90	94.21	5.7	-3.2	35	179

Fanuc five-point type with $\tilde{S}_A = 1.3\tilde{S}_A^*$ and $\tilde{S}_E = 0.8\tilde{S}_E^*$ ($S_r = 1.63 S_r^* = 1.36$) [**4**]

d_0	d_A	d_E	$\frac{L_1}{L_2}$	$\frac{L_4}{L_2}$	γ_C (deg)	L_3	$\frac{F_{o,\max}^{g}}{F_{o,\max}^{*}}$ (%)	$\frac{F_o^0}{F_{o,\max}^{*}}$ (%)	ΔT_h	Δd_v	d_{ch}
$F_o^0 \big/ F_{o,\max}^{*} \le 3.1\%$											
The result has the necessary maximum thrust exceeding the original one by 3.57%.											
$F_o^0 \big/ F_{o,\max}^{*} \le 6\%$											
265	35	80	1.02	0.76	20.34	119.17	91.62	5.7	-2.4	56.1	214

$L_1 + L_2 = 395$ mm, $\tilde{F}_c = 539$ kN, $F_{o,\max}^{*} = 16$ kN, $T_h^* = 1250$ mm, $d_v^* = 277$ mm, $\tilde{\phi} = 92$ for the five-point type