Design of the Swinging-Block and Turning-Block Mechanism with Special Reference to the Mechanical Advantage*

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This study describes a novel and complete solution for turning-block and swinging-block design with particular reference to the maximal average mechanical advantage over a specific swing angle span of the output link. The method yields an optimal set of link lengths for the turning-block and the swinging-block mechanisms. The torque was integrated with respect to the crank swing angle that yields a transmission energy form to maximize the average mechanical advantage. The optimal design solution is determined from the stationary value of the transmission energy. Results of this study can provide a valuable reference for efforts to achieve the high precision and high torque required in the mechanical design.

Key Words: Turning-Block, Swinging-Block, Transmission Angle, Mechanical Advantage

1. Introduction

The swinging-block and turning block mechanism are extensively applied in several mechanical fields. Examples of its use include the recent version of the freight truck loading mechanism, the camera variable zoom lens hood, and others[1]-[9]. Its primary advantage, the strong output torque, is generated by the conversion of a linear force into rotation, as often used in engine mechanisms, such as the oscillating-cylinder engine mechanism, depicted in Fig. 1[10]. Furthermore, following the progress in motor technology in recent years, the high-torque and high-accuracy drive of the micro step-motor has been developed. The design that uses the swinging-block and turning-block mechanism with a confined output rotation angle, normally under $\pi/2$, depends on a high reduction ratio, such as 300 : 1 to support high-precision positioning. Nevertheless, a very small backlash is also required. The important limitation of mecha-

nism is that the relationship between input and output angle is nonlinear. The transmission angle, which determines the mechanical advantage, may vary over a wide range so that the effective torque transmitted to the output link is variable. The mechanical advantage of a particular dimensional design must then be studied. The transmission angle optimizations for the drag link, the crank-and-rocker, and the four-bar linkage are derived in Refs.(11) — (16). However, none of these studies are directly applicable to the swinging-block or turning-block mechanisms.

2. Nomenclature

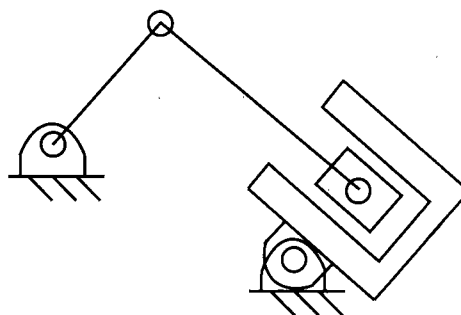L : ground link length
R : crank link length

Fig. 1   Oscillating-cylinder engine mechanism

S : displacement of prismatic joint

S_{max} : maximal displacement of prismatic joint

S_{min} : minimal displacement of prismatic joint

ε : 1/2 specified crank link swinging-angle range

θ : angle of crank link with fixed link

θ_0 : middle-angle of the swinging part of the crank angle θ

ϕ : angle of crank link with linear actuator

α : angle of linear actuator with fixed link

E : average mechanical-advantage

T : output torque transmitted to the output link

F_i : input force from the linear actuator

F_0 : output force from the linear actuator

η : minimum mechanical advantage

3. Swinging-Block and Turning-Block Mechanisms

Figure 2 presents kinematic inversions associated with various selections of ground link of the RRRP kinematic chain[10]. Both the swinging block and the turning block are much less well-known than the slider-and-crank from the same RRRP family, since the output cylinder must swing, which raises manufacturing difficulties. However, current technological advances of the linear motor and the helical motor have greatly simplified the swinging cylinder, such as in the sensor pedestal design, shown in Fig. 3.

The typical design of such a rotational control member may involve a gear head with the servomotor, which suffers from excessive weight and backlash problems. The alternative design adopts a direct-drive motor, which however, requires a higher electric power than specified. There the design of the turning block mechanism is well conceived since it exhibits a low weight-to-power ratio.

Figure 4 illustrates the kinematic structures of the swinging-block and the turning-block mechanisms. L represents the length of the ground-link. R represents the length of the output-link. The slider-
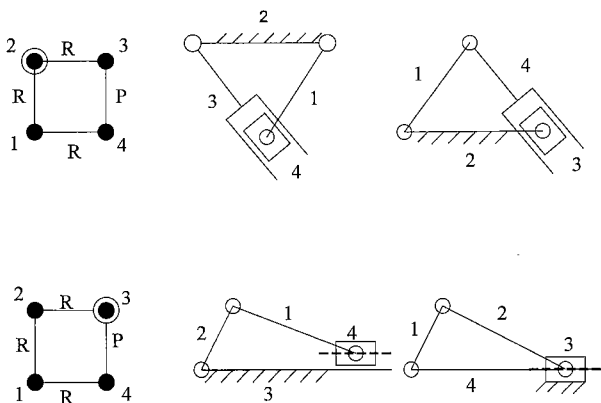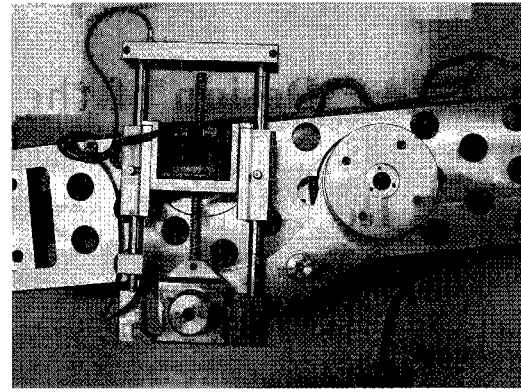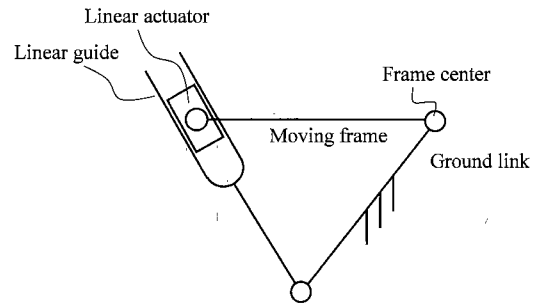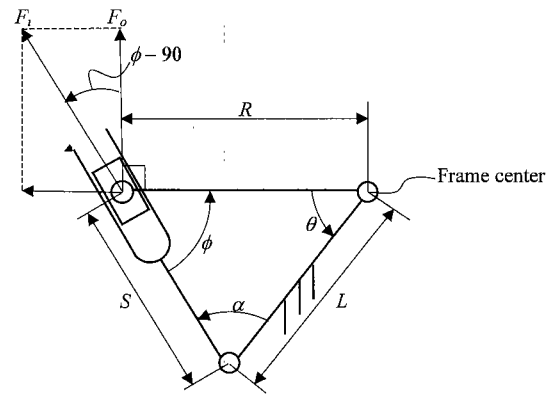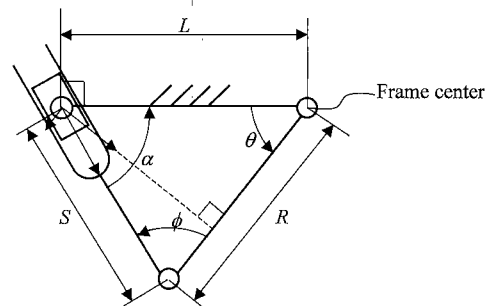

(a)

Fig. 3 (a) Photograph and (b) Kinematic structure of the sensor pedestal mechanism

(a)

(b)

Fig. 4 Force transmitted and kinematic structure of (a) Turning-block and (b) Swinging-block mechanism

Fig. 2 RRRP kinematics inversion

link, depending on the distance traveled by the motor, has a variable length S. With a single degree of freedom, either the swinging block or the turning block mechanism is driven by the input variable S.

The simple trigonometric relation determines the input variable S as a function of the internal angles θ and ϕ as follows.

$$S = \frac{L \cdot \sin \theta}{\sin \phi} \qquad (1)$$

$$S \cdot \cos \phi + L \cdot \cos \theta = R \qquad (2)$$

The cosine law yields,

$$S^2 = R^2 + L^2 - 2RL \cdot \cos \theta \qquad (3)$$

The internal angle ϕ is known as the transmission angle of the swinging block or the turning block mechanism. In Fig. 4, F_i represents the input force, exerted from the linear actuator. T represents the output torque transmitted to the output link, and is a function of the transmission angle, as follows.

$$T = F_0 \cdot R$$
$$= F_i \cdot \sin \phi \cdot R \qquad (4)$$

The mechanical advantage is maximum only when $\phi = \pm 90°$, discouraging the use of swinging block or turning block mechanism to beyond its positions of singularity, $\phi = 180°$.

4. Maximum Average Mechanical Advantage

In practice, the swing angle of the output link is specified in the design of the swinging block or turning block mechanism; that is, on the range of θ is specified. A set of dimensions of the swinging block or the turning block mechanism must be determined to optimize the mechanical advantage over the specified range $\pm \varepsilon$, about the middle-angle θ_0 of the swing angle θ, as shown in Fig. 5. Its application must be limited to $\varepsilon < 90°$ to avoid the singularity. For example, the turning block mechanism may be required to function over a range of swing angle of $\varepsilon = 25°$, a typical value for radar applications. Nevertheless, the workspace of the swinging block mechanism must also be considered.

According to Eqs.(1) and (2), the transmission angle ϕ relates to the swing angle θ as follows.

$$\phi = \tan^{-1}\left(\frac{L \cdot \sin \theta}{R - L \cdot \cos \theta}\right) \qquad (5)$$

The average mechanical-advantage E could be expressed as the integral form of the torque with respect to the swing angle θ, over a specified range $\pm \varepsilon$ with respect to the designing parameter, the middle angle θ_0, as follows.

$$E = \frac{1}{2\varepsilon} \int_{\theta_0 - \varepsilon}^{\theta_0 + \varepsilon} T d\theta$$
$$= \frac{1}{2\varepsilon} \int_{\theta_0 - \varepsilon}^{\theta_0 + \varepsilon} F_i \cdot R \cdot \sin \phi \, d\theta \qquad (6)$$

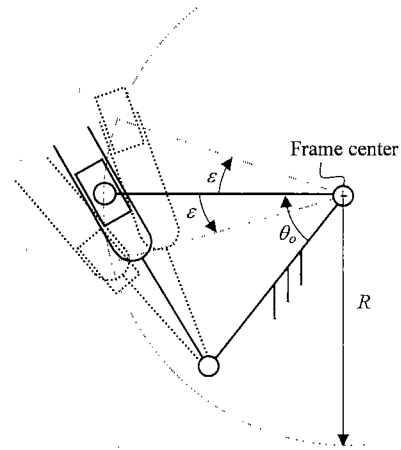Substituting Eq.(5) into the above equation yields,

Fig. 5 Dimensional design for turning block mechanism

$$E = \frac{F_i}{2\varepsilon} \cdot [\sqrt{R^2 + L^2 - 2RL \cos \theta}\,]_{\theta_0 - \varepsilon}^{\theta_0 + \varepsilon} \qquad (7)$$

The maximum average mechanical-advantage with respect to the design parameter θ_0 is obtained by finding the stationary value of the average mechanical-advantage E, as follows.

$$\frac{dE}{d\theta_0} = \frac{F_i}{2\varepsilon} \cdot L \cdot R \left\{\frac{\sin(\theta_0 + \varepsilon)}{S_{max}} - \frac{\sin(\theta_0 - \varepsilon)}{S_{min}}\right\} = 0 \qquad (8)$$

where,

$$S_{max} = \sqrt{R^2 + L^2 - 2RL \cdot \cos(\theta_0 + \varepsilon)} \qquad (9.a)$$

$$S_{min} = \sqrt{R^2 + L^2 - 2RL \cdot \cos(\theta_0 - \varepsilon)} \qquad (9.b)$$

According to Eq.(3), S_{min} and S_{max} are actually the minimum and maximum length extended by the linear actuator, respectively.

In general cases, over a finite swing range ε, the input force F_i, the ground link length L, and the output link length R, must all be non-zero, Eq.(8), yielding,

$$\left(\cos \theta_0 - \frac{R}{L} \cos \varepsilon\right)\left(\cos \theta_0 - \frac{L}{R} \cos \varepsilon\right) = 0 \qquad (10)$$

The optimal solution for θ_0 is obtained, yielding the maximum average mechanical-advantage as,

$$\cos \theta_0 = \frac{R}{L} \cos \varepsilon \qquad (11.a)$$

or,

$$\cos \theta_0 = \frac{L}{R} \cos \varepsilon \qquad (11.b)$$

The second derivative of Eq.(7), which equals the first derivative of Eq.(10) multiplied by a constant k, may be written as,

$$\frac{d^2 E}{d\theta_0^2} = k \cdot H \cdot \sin \theta_0 \qquad (12)$$

where,

$$k = \frac{F_i}{2\varepsilon} \cdot L \cdot R > 0$$

and,

$$H = 8 \cdot \frac{R}{L} \cdot \cos \theta_0 - 4 \cdot \cos \varepsilon \left(1 + \left(\frac{R}{L}\right)^2\right) \qquad (13)$$

The middle-angle θ_0 is selected to be positive, as shown in Fig. 5; such that $\sin\theta_0 > 0$. Hence the sign of Eq. (12) depends on the sign of $H(\theta_0)$. For a design free of any singularity, $\varepsilon < 90°$, that is $\cos\varepsilon > 0$, is required. Substituting Eq. (11.a) into Eq. (13), yields,

$$H = 4\left(\left(\frac{R}{L}\right)^2 - 1\right)\cdot\cos\varepsilon \qquad (14.\text{a})$$

However, substituting Eq. (11.b) into Eq. (13), yields,

$$H = 4\left(1 - \left(\frac{R}{L}\right)^2\right)\cdot\cos\varepsilon \qquad (14.\text{b})$$

The maximum value is obtained from either Eq. (11.a) or Eq. (11.b) by setting $H < 0$, which parameter depends on the R/L ratio of the design. That is, the result of Eq. (11.a) for $L > R$ and that of Eq. (11.b) for $R > L$ are used to obtain the maximum value.

5. Optimal Design

Assuming no energy loss due to friction in the joints or any other viscous damping, the energy output to the output link equals the energy input from the linear actuator, since the total energy is conserved. That is,

$$\int_{S_{min}}^{S_{max}} F_i ds = \int_{\theta_0-\varepsilon}^{\theta_0+\varepsilon} T d\theta \qquad (15)$$

where S_{min} and S_{max} are defined in Eqs. (9.a) and (9.b), respectively. Given a constant input F_i, Eq. (15) may be reformulated as follows.

$$E = \frac{F_i}{2\varepsilon}D \qquad (16)$$

where D denotes the required travel span of the linear actuator,

$$D = S_{max} - S_{min}$$

E is the average mechanical advantage, which was previously defined in Eq. (6). Equation (16) shows that the average mechanical advantage E is proportional to the distance traveled by the linear actuator D. Therefore, a linear actuator that can be extended farther is always preferred for its greater mechanical advantage.

According to Eq. (1), the transmission angles ϕ_{min} and ϕ_{max} are defined as follows.

$$\sin\phi_{min} = \frac{L\cdot\sin(\theta_0+\varepsilon)}{S_{max}} \qquad (17.\text{a})$$

$$\sin\phi_{max} = \frac{L\cdot\sin(\theta_0-\varepsilon)}{S_{min}} \qquad (17.\text{b})$$

Where θ_0 fulfills one of Eqs. (11.a) and (11.b) to yield the maximum average mechanical advantage. According to Eqs. (8), (17.a) and (17.b),

$$\sin\phi_{min} = \sin\phi_{max} \qquad (18)$$

Thus, the travel span of the linear actuator D can be related to the link length L, as follows.

$$D = \frac{L}{\eta}\cdot\{\sin(\theta_0+\varepsilon) - \sin(\theta_0-\varepsilon)\} \qquad (19)$$

where η represents the minimum mechanical advantage and,

$$\eta = \sin\phi_{min} = \frac{T}{F_0\cdot R}$$

The design problem concerns five design parameters, D, L, R, η and ε. These five design parameters uniquely determine the four turning block mechanism link lengths and one the middle swing-angle. Of the five design parameters, the swing angle span ε is provided as a design specification. This set of design parameters can again by normalizing the link length, for example by letting $R = 1$. Consequently, only three normalized design parameters are to be determined; they are η, D/R and L/R.

Optimal design problems may be separated into two categories. The first is for $L \geq R$. Equation (11.a) is applied to find the optimal solution. Manipulating and simplifying Eqs. (17.a), (11.a) and (9.a) yield,

$$\eta = \cos\varepsilon \qquad (20)$$

For $L \geq R$, the design parameters η and ε are not independent. In design procedure, η can not be freely specified for a given swing span ε. Substituting Eq. (11.a) into Eq. (19) and combining with Eq. (20) yields,

$$\frac{D}{R} = 2\cdot\sin\varepsilon \qquad (21)$$

The second category of problem has $L < R$. Equation (11.b) is applied to find the optimal solution. Manipulating and simplifying Eqs. (17.b), (11.b) and (9.b) yields,

$$\text{for } R\cdot\sin\theta_0 > L\cdot\sin\varepsilon: \ \eta = \frac{L}{R}\cdot\cos\varepsilon \qquad (22.\text{a})$$

$$\text{for } R\cdot\sin\theta_0 < L\cdot\sin\varepsilon: \ \eta = \frac{L}{R}\cdot\cos\varepsilon \qquad (22.\text{b})$$

Since Eqs. (22.a) and (22.b) contradicts the condition that $L < R$, no optimal solution exists. The second category is discarded because of the need to obtain a good mechanical advantage.

6. Design Procedure

The design procedure is summarized as follows.
Step 1: Specify ε.
Step 2: Set the L/R ratio to no less than 1.
Step 3: Obtain η and the D/R ratio from Eqs. (20) and (21), respectively.

For the configuration illustrated in Fig. 6, $R = 1$, $L = 2$ and $\varepsilon = 30°$ are set. The optimal value of the design parameter θ_0 is obtained from Eq. (11.a), yielding $\theta_0 = 64.34°$ and $D/R = 1$. Figure 7 presents more general cases subjected to different ε versus D/R and ε versus η. Figure 8 plots the curved surface of ε and R/L versus θ_0.

7. Conclusion

This investigation has completely solved the design optimization problem subjected to constant input force for the general turning-block as well as
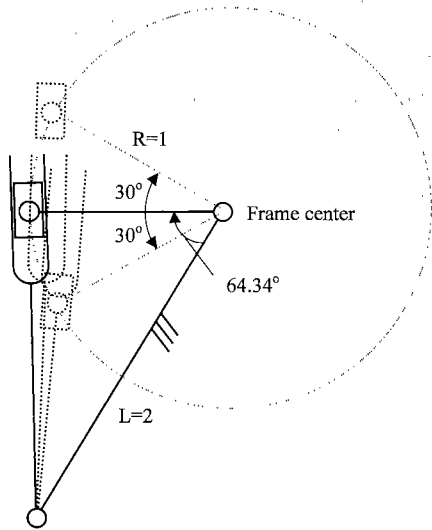
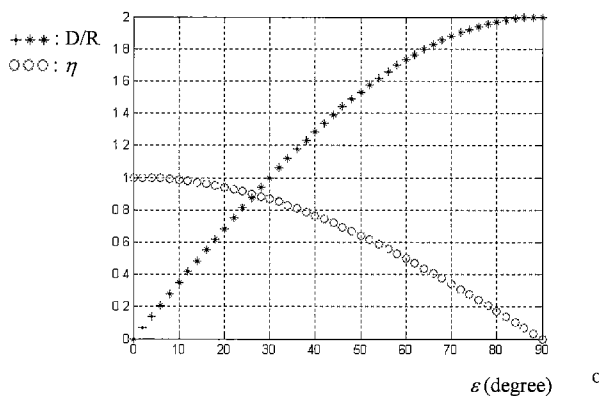Fig. 6   Example of turning-block illustrated configuration

tage with the swing angle span ε of the output link. Nevertheless, the workspace associated with the L/R ratio is submitted to the designer in advance, and the optimal design procedure need not be further verified. The D/R ratio, which affects the assessed cost of the linear actuator, can be easily determined from the given swing span ε. Additional multi-objective optimization of the total cost-performance for different L/R ratios, and ε values, may be performed for different applications in the future.

Acknowledgements

Fig. 7   ε versus D/R and ε versus η

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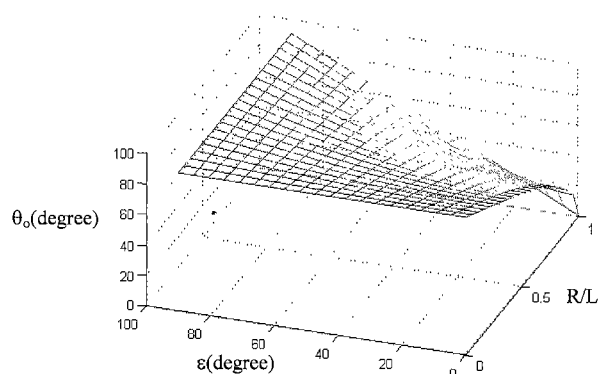(13)   Freudenstein, F., Designing Crank and Rocker Links with Optimum Force Transmission, Prod-

Fig. 8   ε versus R/L versus $θ_0$

swinging-block mechanisms with a special reference to the average mechanical advantage. The results were given for two types of applications, $L>R$ and $L<R$. No optimal solution exists in the $L<R$ category. The rule-of-thumb design procedures allow the engineer to correlate the optimal mechanical advan-

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