A Dynamic Analysis of an Adjustable Conveying Machine*

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In the previous paper, the synthesis of a planar adjustable 9-link mechanism was reported, as a practical example of an automatic conveying machine which generates an L-path. It has been found that the driving system of the above-mentioned machine, which is constructed of an analog computer, an induction motor, a magnetic power clutch and brake, is so complicated that the system made the response worse, and that positioning errors of the machine hand at the edge points of the path increased. In this paper, a 12-link adjustable mechanism is proposed, which is made by joining a 4-link mechanism to the input link of the above 9-link mechanism, for the purpose of making the crank motion uniform in one direction and simplifying the electric controller of the driving system. Moreover, an optimum distributed mass and moment of inertia of the input link is determined so as to minimize the input torque. The dynamic characteristics of the machine are also investigated theoretically and experimentally. As a result, the velocity response and the positioning accuracy of the machine hand at the edge points of the path can be greatly improved.

Key Words: Multi-Link Mechanism, Path Generator, Machine Hand, Balancing Device, Positioning Accuracy

1. Introduction

The authors have discussed the adjustable mechanism that is applicable to automatic conveying machines in previous reports^{(1)~(4)}. In these studies, however, conveying machines equipped with adjustable mechanisms were likely to give rise to a delayed response in motion and, in consequence, a less accurate positioning at the edge points of the conveying path.

The cause of these insufficient results was apparently due to the limitations of the built-in electric control system, including the brake and clutch components, which was structurally complicated, for governing stopping and reverse motions. Consequently, the conveying speed often failed to pinpoint stopping, although it had to synchronize with the initiation or termination point of time, respectively. During a low

speed performance, moreover, a significant difference was recognized in the velocity, compared with the ideally intended speed. Because of these problems, the necessary level of brake-controlling voltage was inevitably increasing, as was the brake load.

In this study, therefore, we attempt to investigate these problems under the following circumstances: the reciprocating motion of a driving link is converted into uniform revolutions in one direction, and machine hand motions at each edge point are controlled to assure good response and accurate positioning, as maintained through a mechanically driven control system in which a simple electric circuit is built-in. The control system, simultaneously, saves energy consumption in the power used for braking, particularly during periods of low speed performance. In order to realize an optimal, practical model of an adjustable conveying machine, its static and dynamic characteristics are further analyzed, with particular regard to ensuring that it can achieve both a decrease in input torque variance and an increase in response capability.

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2. Synthesis of Conveying Mechanism

A planar adjustable 9-link conveying machine⁽²⁾, diagrammed in Fig. 1, is synthesized to satisfy the requirements of the set L-path; that is, a machine hand holds a packet of the conveying object at the initiation point A_k , and after passing through the point B_k , the machine hand unloads it at the termination point C_k . Subsequently, the machine hand that is already empty at this point immediately turns back as the reverse revolution of the link e begins, and returns to the point A_k , again. This mechanism begins to move through a driving system⁽²⁾, as illustrated in Fig. 2, which is composed of various components, such as an induction motor, normal and reverse revolution clutches, and a brake unit. Since the position of the pair point J_{an} is arbitrarily adjustable within a range of points from 0 to 5, the path can readily satisfy the required elements of $A_k B_k C_k$ ($k=0$ to 5). As plotted in Fig. 3, changes in the angular velocity of the link e are recorded at each driving cycle when the link e has

Fig. 1 Planar adjustable 9-link conveying machine

Fig. 2 Driving system of 9-link machine

revolved from the initiation of response to termination. In the diagram, the results are then divided into three regions, such as starting period PQ, constant driving period QR_j ($j=1$ to 4), and braking period R_jS . In this model, machine motions were controlled by an analog computer, which enabled us to analyze the input data concerning the differences taking place between the actual measured speed and the expected speeds in the QR_j and R_jS , with the latter two representing the ideal velocity during a constant driving period and a braking period, respectively. However, the machine's electric control system was complex in structure; that is, it was comprised of those components which can act as the following units: a converter, which converts measurements of the established parameters for velocity into electric signals; a detector, which measures a variety of control volumes and converts them into electric signals; a processor, which calculates and sends control signals based on the established values and detector signals; and an amplifier, which can amplify the amplitude of such control signals. Accordingly, input signals of the machine were likely to give rise to time lags in the corresponding output signals so that the actual conveying speed response curves represented with thick lines in Fig. 3 did not match with the ideal curves represented by thin lines. Particularly, because the end speed of link e was unable to pinpoint stopping at the termination point S, an acute braking performance was needed to avoid an accident, where the driving link could brush against a standstill pair axle. During the braking period R_jS , in Fig. 3, the level of brake-controlling voltage increased proportionately as the degree of difference became larger between the actual speed and the ideal speed, and the brake perfor-

Fig. 3 Velocity diagrams: Optimal and actual conveying speeds of 9-link machine

mance load likewise became heavier.

In this paper, therefore, we propose a plan for a planar adjustable 12-link conveying machine which is incorporated with a set of three additional links..... the links b, c, and d, each having a corresponding gear, as shown in Fig. 4.....and then synthesize its mechanism. In Fig. 4, uniform revolution in one direction is transmitted directly to a driving shaft from a motor. A process of optimization⁽⁶⁾ is carried out using a three-point synthesis method⁽⁵⁾ that allows us to transmit oscillating motions to the link e through a newly added 4-link mechanism satisfying the required conditions of the path. Optimal dimensions for the length of each link are investigated. In this case, it is absolutely necessary to establish a gear ratio of 9 for the gear d, and 2 for the gear e. Thus, the machine's input and output angular displacement functions θ' and φ' , and the angular velocity $\dot{\theta}$ are represented as follows:

$$R_1 \cos \theta' - R_2 \cos \varphi' + R_3 = \cos(\theta' - \varphi') \quad (1)$$

$$\theta' = 340.8 - \theta, \varphi' = 91.3 - 2/9\theta \quad (2)$$

$$\dot{\theta} = 9/2 \{ \dot{\theta}' l_b \sin(\theta' - \theta_e) / l_d \sin(\theta_e - \theta_d) \}$$

Then,

$$\theta_e = \sin^{-1} \{ -l_b \sin \theta' - l_d \sin \theta_d \} / l_c$$

$$\theta_d = \cos^{-1} (R_1 \cos \theta' + R_3) / \sqrt{1 + 2R_2 \cos \theta' + R_2^2} + \tan^{-1} \{ \sin \theta' / (\cos \theta' + R_2) \} + \pi$$

$$R_1 = l_a / l_d, R_2 = l_a / l_b$$

Fig. 4 Planar adjustable 12-link conveying machine

$$R_3 = (l_a^2 + l_b^2 - l_c^2 + l_d^2) / 2l_b l_d$$

are obtained, where the notations l_a , l_b , l_c and l_d represent the respective link length, and θ the angular displacement of the link e. Table 1 shows measurements obtained from each dimension or geometric constant of the mechanism.

Instead of the analog computer used in the previous model, a digital computer is used in this model to control the mechanism's driving system, as illustrated in Fig. 5. The computer can initiate brake action..... the brake is connected directly to the input shaft..... immediately before points where the machine hand meets at the initiation and termination edges, based on the timing of the information stored in its memory. It can also control the positioning of the machine hand stop, indicating disconnection of clutch action at the edge points of the path. In this driving system, moreover, the link e can directly follow oscillating motions, and motions are determined much more accurately through the support of the newly added three links so that only a simple, single clutch plus a brake component can readily provide enough action so that the machine hand can stop transiently, but accurately, at each edge point of the path. In the previous model, however, two sets of clutches were needed to separately share the roles of normal and reverse revolution. Thus, it is asserted that this achievement simplifies

Table 1 Dimensions of experimental mechanism

Link notation	Length mm	Link notation	Length mm	Name of angle	Angle
a	170.00	c	152.00	θ_{a0}	47.00°
a ₀	305.51	d	70.00	θ_{a1}	43.87°
a ₁	321.95	e, j	150.00	θ_{a2}	41.09°
a ₂	342.51	f	226.78	θ_{a3}	50.45°
a ₃	292.32	h	303.62	θ_{a4}	54.07°
a ₄	280.78	i, l, n, u	75.00	θ_{a5}	57.73°
a ₅	269.30	e', j'	50.00	θ_f	18.00°
b	33.33	k, f'	420.00

Fig. 5 Driving system of 12-link machine

the mode of control, compared with that of the previous model.

3. Theoretical Analysis on the Kinetics of Conveying Machines

3.1 Dynamic analysis of planar multi-link conveying machines

In the model of the planar adjustable 12-link conveying machine shown in Fig. 4, input torque is transmitted to a 9-link mechanism by the link e, whose revolving performance can be adjusted through

Table 2 Kinematic constants of the mechanism

Link notation	Mass kg	Moment of Inertia kg·m ²	Position of center of gravity	
			p mm	q mm
b	1.2100	6.946×10 ⁻³	10.6	0.0
c	0.6227	1.845×10 ⁻³	76.0	0.0
d	5.0400	5.018×10 ⁻²	4.0	0.0
e	0.1785	2.932×10 ⁻³	45.4	16.7
f	0.4047	2.489×10 ⁻²	210.7	34.0
h	0.1830	6.356×10 ⁻³	150.6	0.0
j	0.1926	2.203×10 ⁻³	74.7	3.7
k	0.2984	1.833×10 ⁻²	200.4	0.0
l	0.6321	1.352×10 ⁻²	165.8	45.8
n	0.1100	3.352×10 ⁻⁴	35.8	8.8
u	0.1010	3.832×10 ⁻⁴	48.0	0.0
v			250.0	32.0

Fig. 6 Fluctuation of the torque affecting link e of the 9-link machine at each adjusted position ($k=0$ to 5)

the gear action of the other, newly added 4-link mechanism. The magnitude of the torque is represented by the equation described below, which has been derived from the equation of motion for the link e.

$$T_e = \{I_e + m_e(P_e^2 + q_e^2)\}\ddot{\theta} + F_{xe}l_e \sin \theta - F_{ye}l_e \cos \theta + (P_e \cos \theta - q_e \sin \theta)m_e g \quad (3)$$

Then, the force of action of the pair point $J_{ex}(F_{xe}, F_{ye})$ is given by the solution of the equation of motion that involves the rest of the 9-link mechanism, excluding the link e. The force is denoted as follows:

$$F_{xe} = F(m_i, \ddot{x}_i, \ddot{y}_i, l_i, l'_i, p_i, q_i, I_i, \ddot{\theta})$$

$$F_{ye} = F(m_i, \ddot{x}_i, \ddot{y}_i, l_i, l'_i, p_i, q_i, I_i, \ddot{\theta})$$

("i"s represent the notation of each link.)

In addition, the angular acceleration $\ddot{\theta}$ is determined through the differentiation of Eq. (2).

Linguistic notations used are defined as follows: I_e and I_i are the respective moments of inertia when links revolve around the center of gravity; m_e and m_i are the respective masses; p_i and q_i are the distances between the pair axle and the center of gravity, with the former being parallel to each link length, and the latter at right angles; $\ddot{x}_i$ and $\ddot{y}_i$ are the accelerations at the center of gravity; g is the acceleration of gravity; and l_i is the length of the link. The angular velocity of the link e is postulated from four different diagrams of ideal velocity, as shown in Fig. 3. Then, the torque T_e , which affects link e, is determined from the Eq. (3) by using the

Fig. 7 Fluctuation of the torque affecting link e of the 12-link machine at each adjusted position ($k=0$ to 5)

kinetic constants listed in Table 2. Figure 6 shows the torque of the link e at each of the adjusted positions ($k=0$ to 5) that corresponds to the given ideal velocity diagram PQR,S (driving cycle : 5 sec) . If the torque T_e that affects link e is determined at the angular velocity $\dot{\theta}=12\text{rpm}$ of the link b, the solution gives a curve, such as plotted with a chain line in Fig. 7. In consequence, it is shown that these two systems revealed almost all equivalent curves and magnitudes, regardless of the driving mode, and that the input torque of each system, likewise, affected considerably at and around the oscillating angles of $\Theta=0(\theta=\pi)$ and $\Theta=\pi(\theta=5/3\pi)$ rad.

3.2 The improvement of kinetics in the conveying machine

The parameters, as illustrated in Fig. 8, where m is the balanced mass of the link e, I is its moment of inertia about the center of gravity, and p and q are the relative positions of the center of gravity, are analyzed in Figs. 6 and 7. Consequently, the torque T that occurs at the pair point J_{ae} can be represented as follows :

$$T = -\{I + (p^2 + q^2)m\}\ddot{\theta} + (p \cos \Theta + q \sin \Theta)mg \quad (4)$$

If calculations are carried out to determine each of the given parameters, such as m , I , p and q , by applying a method of direct search to Eqs. (3) and (4), in order to establish the level of minimum torque which affects the link e, respective optimal values can be obtained, as follows : $m=2.0\text{ kg}$, $I=1.720 \times 10^{-2}\text{ kg}\cdot\text{m}^2$, $p=86.6\text{ mm}$, and $q=14.0\text{ mm}$. Subsequently, if the torque T_M affecting the link e in the 12-link conveying machine has the above-mentioned values as its mass and moment of inertia, it can be demonstrated that its changes can be represented as the curves of a continuous line, as shown in Fig. 7. As shown by the given figure, it is apparent that, if the machine can be arranged so as to maintain an optimal state of kinetic balance for every conveying path, the torque can theoretically lower its maximum values by up to

Fig. 8 Counterweight

approximately 50%, compared with that of the pre-balanced state. In addition, there can also be obtained a minimum torque variance throughout every portion of the entire path.

4. Modeling and Experiments for the Conveying Machine

4.1 Experimental apparatus and method

The experimental apparatus, as demonstrated in Fig. 9, is assembled with an adjustable conveying machine, as characterized in Fig. 4 and Table 1, and a driving system, as illustrated in Fig. 5. A solenoid-type on-off system is adopted for the machine hand mechanism to reduce working wear as far as possible. The magnitude of torque is measured with strain gauges that are attached to the pair shaft J_{ae} , while the angular displacement and velocity are detected by a potentiometer and a tachogenerator, respectively. Further, using a non-touch-type displacement sensor, each location of the stop position of the machine hand is measured, and positioning errors are experimentally determined.

4.2 Experimental results and discussions

4.2.1 Conveying speed and energy consumption for braking

Figure 10 shows the relation between the angular displacement θ of the driving link and the angular velocity $\dot{\theta}$ of the link e in the case where the pre-

Fig. 9 Experimental apparatus of 12-link machine

balanced or unbalanced planar 12-link conveying machine has actual values of 0.5 kg for the mass of the conveying object and of $\dot{\theta}=12$ rpm, and which corresponds to each available path ($k=0$ to 5). For the angular displacement of the driving link, as deduced from the given figure, all experimental values obtained, regardless of path, differ from the corresponding theoretical values under conditions at $2/3\pi$ and between $4/3\pi$ and $3/2\pi$ rad. These phenomenon are caused by the fact that the machine hand generates an acceleration of larger magnitude during the period when it turns in a direction towards the x -axis of the set L-path from the Y -axis, and the driving link, in consequence, fails to properly maintain its uniform motion.

Subsequently, the diagram of velocity was viewed under the same conditions after the mechanism was adjusted to maintain its kinetic balance, and the results are shown in Fig. 11.

In this figure, the variance of experimental values obtained decreases for each path, compared with that of Fig. 10, and both experimental and theoretical values match sufficiently.

Further, a comparison of the energy consumption for braking was studied between the 9-link model, described in the previous paper⁽²⁾, and this 12-link model under the same experimental conditions where both machines convey the mass of 0.5kg, respectively. Then, levels of the brake-controlling voltage responsible for machine hand stop were measured at initiation and termination points for both machines. As a result, it is suggested that the 12-link machine can almost always maintain a constant power level of 2J during a duration where its driving cycle falls within a range of 3 to 5 seconds, while the power level for the 9-link machine fluctuated over a larger range, from

approximately 11 to 23J.

Based on these results, it is apparent that the energy consumption level can be lowered more in the 12-link conveying machine than in the 9-link machine. This means that brake performance is improved in the former, where its brake acts only to manage the necessary stop of the machine hands at each edge point. In the latter case, however, the brake acts throughout the low speed performance, since the driving mode has been altered to the mechanical system with simple, electric controlling equipment, as shown in Fig. 5, from the complicated, electric control system discussed in Section 2 of this report.

4.2.2 Fluctuation of torque

Experimental values of the torque affecting the link e are shown with dotted lines in Figs. 12(a) and (b) for the case where the planar 12-link conveying machine is in state of unbalance, the values are approximately the same as the corresponding theoretical values represented by the continuous lines. However, the experimental values also show a tendency that fluctuations of minor significance occur at and around the state of $\theta=3/4\pi$ rad. These phenomena are caused by the fact that the acceleration of the machine hand reinforces the x -axis direction of the path at the place where the machine hand turns from the y -axis to the x -axis, and in consequence, the acceleration may act as another driving force factor. In the case where the machine has been balanced, on the other hand, experimental values, as shown with the dotted lines in Figs. 13(a) and (b), are almost all the same as the corresponding theoretical values, and a decrease of the torque variance is apparent.

4.2.3 Accuracy of positioning

First, respective target values were established as the machine hand position under the conditions where

Fig. 10 Changes in the conveying velocity at each adjusted position ($k=0$ to 5) in the unbalanced 12-link machine (at $\dot{\theta}=12$ rpm)

Fig. 11 Changes in the conveying velocity at each adjusted position ($k=0$ to 5) in the balanced 12-link machine (at $\dot{\theta}=12$ rpm)

the link e had input angles of $\theta=0$ and $3/2\pi$ rad in the above-mentioned 9-link conveying machine, and as the edge point positions of each machine hand in the 12-link machine. Then, each standstill position was

measured at the edge point after operating the machine for a driving cycle time ranging from 3 to 5 seconds, and each difference from the established target values was measured as a positioning error.

(a) the torque affecting link e (at $\dot{\theta}=12$ rpm) at each adjusted position ($k=0$ to 5)

(b) the torque responding to changes of the velocity at each adjusted position ($k=0$ to 5)

Fig. 12 Fluctuations of the torque in the unbalanced 12-link machine

(a) the torque affecting link e (at $\dot{\theta}=12$ rpm) at each adjusted position ($k=0$ to 5)

(b) the torque responding to changes in the velocity at each adjusted position ($k=0$ to 5)

Fig. 13 Fluctuations of the torque in the balanced 12-link machine

(a) 9-link machine

(b) 12-link machine

Fig. 14 Standard deviations of the machine hand positioning at edge points

Average values of the errors were, approximately, 0.1 mm for the x -axis direction at both the initiation and termination points, and 0.05 mm for the y -axis direction in the 9-link machine, while the error was about 0.1 mm for both the x -axis and y -axis directions in the 12-link machine. Standard deviations obtained from the mean of the respective measurements are shown in Fig. 14. As a result, the 12-link machine can maintain a superior, reproducible accuracy with less deviation, compared with its counterpart. This means that the driving mechanism of the 12-link machine can readily control the machine hand performance as a result of its adoption of the alternative system, noted above. Hence, changes in the angular displacement θ of the link e to the input angle θ can be structurally minimized around both the edge points.

5. Conclusion

In place of the planar adjustable 9-link conveying machine, as described in the previous report, a model of a planar adjustable 12-link conveying machine with an alternate driving system was proposed for synthesis and modeling, and the results of theoretical and experimental analyses pertaining to its kinetics are elucidated as follows:

(1) In the 12-link machine, non-uniform motion of the driving link was converted into uniform revolutions in one direction using an additional 4-link mechanism, while in the 9-link machine, its driving speed was controlled by an electric mode of control, which included clutch and brake components. Due to the simplification of the control system, a portion of the performance dependent on electric speed control-

ling significantly decreased, and in consequence, the machine's response capability improved.

(2) Brake performance, in particular, was always active throughout the low speed operations in the 9-link machine. In the 12-link machine, however, the role of brake performance was limited to only maintaining an accurate positioning of the machine hand at both the edge points so that the energy consumption for braking was significantly decreased.

(3) Viewed from the results which have been obtained from the static and dynamic analyses of the given 12-link machine, it was obvious that the maximum magnitude of the torque affecting the link e will reduce up to approximately one half the level of that which was measured at a state of unbalance.

(4) Based on the analytical results regarding the kinetics of the machine, it was recognized that the accuracy of the positioning of the machine hands can be considerably improved at the initiation and termination points during the conveying.

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