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THEORETICAL ANALYSIS OF DYNAMICS OF FOUR-BAR BEAT-UP MECHANISM OF A LOOM

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Abstract—During the design of a beat-up mechanism, problems arise in its evaluation with respect to the individual parameters. These problems are related, mainly, to the influence of absolute values of the parameters (such as the angle of weft insertion, the swing angle of the sley with the reed, the structure of the mechanism, the space available, etc.) on the correct and smooth functioning of the mechanism as a whole. This paper deals with a method for the rapid and easy study of the effect of various parameters on the dynamic characteristics of the designed mechanism. A mathematical model has been formulated using Lagrange's equation of type II. This model takes into account the mass-related parameters of the mechanism, joints between its members, technological stress and the non-linearity of the effect of this stress in the system because of the design clearance and transfer function of the mechanism.

INTRODUCTION

打纬机构与开口机构共同影响着织机的纬纱质量和生产效率，是织机的关键机构之一。它也影响了机器的动态特性。因此，在织机设计中，织机的结构和设计是非常重要的。用单一的标准来评价击穿机构的性能是不可能的。优化设计的基本标准包括引纬角度、最大加速度和加速度曲线的形状、天空的摆动角度、回转轴的摆动轴的位置、击穿点和机构可用的空间。这些标准的重要性和临界取决于结构、机制和系统的建设和布局的引纬。

The beat-up mechanism together with the shedding mechanism, influences the quality of the pick and productivity of a loom and thus forms one of its critical mechanisms. It also affects the dynamic characteristics of the machine as a whole. Therefore, in designing a loom, much attention is paid to its structure and design. It is not possible to evaluate the performance of a beat-up mechanism by a single criterion. The basic criteria for an optimum design include angle of weft insertion, maximum acceleration and the shape of the acceleration curve, swing angle of the sley, position of the axis of the rocking shaft of the sley, beat-up point and the space available for the mechanism. The importance and the criticality of each of these criterion will depend on the structure, construction and layout of the mechanism and the system of weft insertion.

如果不考虑开口机构，打纬机构就不能单独进行优化。因此，在设计的初始阶段，针对引纬角度对击穿机构进行几何优化是很重要的。一般来说，很难确定每一个标准的相对重要性，其中一些标准会以与其他标准相反的方式影响机制的性能。专家的意见在大多数情况下是不同的，甚至在某些情况下是相反的。

The beat-up mechanism cannot be optimized in isolation without considering the shedding mechanism. Hence, in the initial stages of design it is important to optimize geometrically the beat-up mechanism *vis-à-vis* the angle of weft insertion. Generally, it is difficult to determine the relative importance of each of these criteria, some of which will affect the performance of the mechanism in the opposite manner to others. Expert opinions are, in most cases, different, and may even be contrary in some cases.

本文的目的是要找到一个方法的动态特性和各种参数的影响,他研究了使用一个数学模型。

The aim of this paper is to find a method by which the dynamic characteristics and the influence of the various parameters can be studied using a mathematical model.

MATHEMATICAL MODEL

根据上述要求，建立了四杆机构和六杆机构的数学模型。本文只求解了四连杆机构。这里考虑两种四连杆机构的情况，一种是一侧只有曲柄和连杆，如图!所示，另一种是两侧都有曲柄和连杆，如图2所示。六连杆机构已经在其他地方解决了[1]。

Mathematical models have been formulated for a four-bar linkage and a six-bar linkage mechanism, keeping in view the requirements listed above. In this paper, only the four-bar linkage mechanism is solved. Two cases of the four-bar linkage mechanism are considered here, one with the crank and linkage on only one side, as shown in Fig. 1, and the other with the crank and linkage on both sides, as shown in Fig. 2. Six-bar linkage mechanisms have been solved elsewhere [1].

真正的打纬的机制不同于理想弹性元素中的一个主要运动链，连续分布质量和引入非线性的影响主要是通过运动的间隙对加载的驱动和随机特性的筘座经表。

A real beat-up mechanism differs from an ideal one mainly in the elastic elements in the kinematic chain, the continuously distributed mass and the introduced non-linearity, which are affected mainly by the clearances in the kinematic pairs in the drive and by the random characteristics of loading of the sley by the warp sheet.

这类机构的数学模型应该始终是在所要求的结果的精确性和真实机构的精确性之间的折衷。

A mathematical model of this type of mechanism should always be a compromise between the required accuracy of the result and exactness in representing the real mechanism.

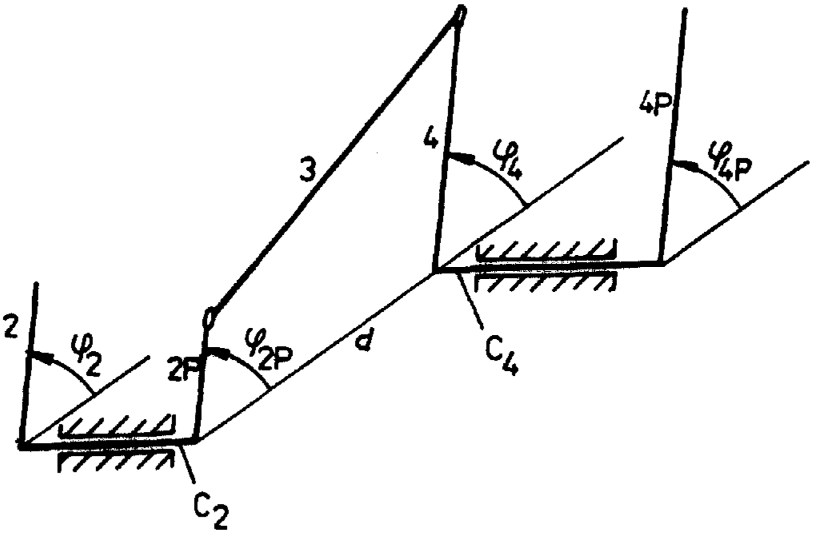

Fig. 1. Crank and linkage on one side only.

The formulated mathematical model is based on the following assumptions:

单元的质量集中在关节上：

(a) the masses of the elements are concentrated on the joints;

(b) all the members are perfectly rigid; 所有构件为完全刚性；

绝对刚性拉杆由具有扭转刚度C的扭转杆相互连接；

(c) absolutely rigid draw-bars are mutually connected by torsional rods with a torsional rigidity C;

(d) the clearance meant for mounting is considered between the draw-bars connected by torsional rods; 用于安装的间隙是考虑由扭转杆连接的拉杆之间的间隙；

(e) the effect of the mass of the members is neglected; and

(f) the complete sley with reed is considered as absolutely rigid, joined elastically to the driving draw-bar. 带筘座的整个筘座被认为是绝对刚性的，弹性地连接到驱动牵引杆上。

In the mathematical model the following Lagrange's equation of type II is used:

$$\frac{\mathrm{d}}{\mathrm{d}t}\left(\frac{\partial K}{\partial \dot{q}_i}\right)-\frac{\partial k}{\partial q_i}=-\frac{\partial U}{\partial q_i}-\frac{\partial R}{\partial \dot{q}_i}, \qquad (1)$$

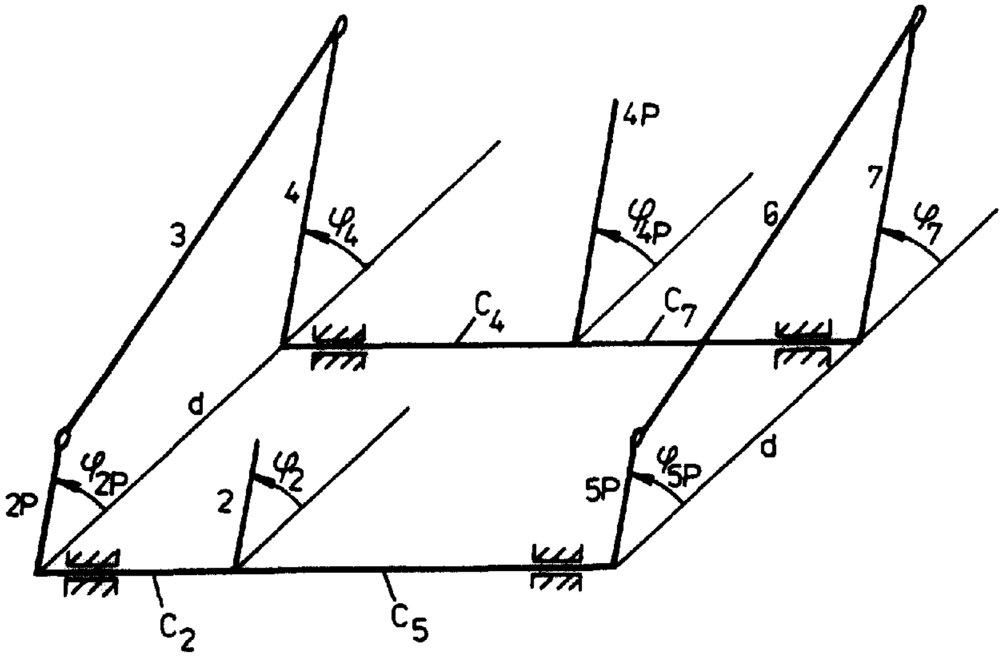

Fig. 2. Crank and linkage on both sides.

where K is the total kinetic energy of the system, U is the total potential energy of the system, R is the dissipative function and q is any general coordinate.

For the case shown in Fig. 1, the kinetic and potential energy and dissipative function for the system can be written as follows:

$$K = \tfrac{1}{2} I_{2P}\dot{\varphi}_{2P}^2 + \tfrac{1}{2} I_4\dot{\varphi}_4^2 + \tfrac{1}{2} I_{4P}\dot{\varphi}_{4P}^2$$

$$U = \tfrac{1}{2} C_2(\varphi_{2P} - \varphi_2)^2 + C_4(\varphi_{4P} - \varphi_4)^2 + \int_0^{\varphi_4} M_4 \, \mathrm{d}\varphi_4$$

$$R = \tfrac{1}{2} k_2(\dot{\varphi}_{2P} - \dot{\varphi}_2)^2 + \tfrac{1}{2} k_4(\dot{\varphi}_{4P} - \dot{\varphi}_4)^2 \tag{2}$$

where I_{2P}, I_4 and I_{4P} are the moments of inertia of individual members; φ_2, φ_{2P}, φ_4 and φ_{4P} are the angles of rotation of individual members; $\dot{\varphi}_2$, $\dot{\varphi}_{2P}$, $\dot{\varphi}_4$ and $\dot{\varphi}_{4P}$ are angular velocities; C_1 and C_2 are the torsional rigidity of the connecting rods of the draw-bars; K_1 and K_2 are the coefficients of damping; and M_4 is the moment effected on the sley by the warp sheet. Relative positions of members $2P$ and 4 are given by the transfer functions of the mechanism in the form

$$\varphi_4 = f_{24}(\varphi_{2P})$$

$$\dot{\varphi}_4 = \mu_{24} \cdot \dot{\varphi}_{2P}$$

$$\ddot{\varphi}_4 = \nu_{24}\dot{\varphi}_{2P}^2 + \mu_{24}\ddot{\varphi}_{2P}. \tag{3}$$

Exact functions as applied to members $2P$, 3 and 4 of a four-bar linkage mechanism have been given in [2].

Substituting relations (2) and (3) in equation (1), the following equations can be obtained, which can be numerically integrated:

$$\frac{\mathrm{d}\varphi_{2P}}{\mathrm{d}t} = \omega_{2P}$$

$$\frac{\mathrm{d}\omega_{2P}}{\mathrm{d}t}(I_{2P} + I_4\mu_{24}^2) = -I_4\mu_{24}\nu_{24}\omega_{2P}^2 - C_2(\varphi_{2P} - \varphi_2) + C_4\mu_{24}(\varphi_{4P} - \varphi_4) - k_2(\omega_{2P} - \omega_2) + k_4\mu_{24}(\omega_{4P} - \omega_4)$$

$$\frac{\mathrm{d}\varphi_{4P}}{\mathrm{d}t} = \omega_{4P}$$

$$\frac{\mathrm{d}\omega_{4P}}{\mathrm{d}t} I_{4P} = -C_4(\varphi_{4P} - \varphi_4) - k_4(\omega_{4P} - \omega_4) - M_4, \tag{4}$$

where ω_{2P}, ω_2, ω_{4P} and ω_4 are the angular velocities of individual members as represented in Fig. 1.

Moment M_4 in the equations of motion of member $4P$ represents the resistance of the fabric to the beat-up, and this can be considered to be in different forms. In this work, this moment is considered to be a sine function in the form (see Fig. 3)

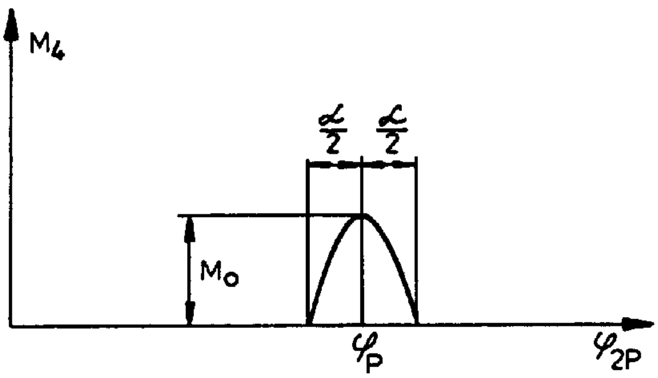

Fig. 3. Loading moment M_4.

$$M_4 = M_0 \sin \frac{\pi\left(\varphi_{2P} - \varphi_P + \frac{\alpha}{2}\right)}{\alpha}. \tag{5}$$

Also, M_4 is considered to act only in the interval

$$2n\pi + \varphi_P - \frac{\alpha}{2} < \varphi_{2P} + 2n\pi < 2n\pi + \varphi_P + \frac{\alpha}{2}, \quad n = 0, 1, 2, \ldots, \tag{6}$$

where angle φ_P is the value of parameter φ_{2P} corresponding to the position of the mechanism at the beat-up given by the relation

$$\varphi_P = \arccos \frac{(l_{2P} + l_3)^2 + d^2 - l_4^2}{2d(l_{2P} + l_3)}. \tag{7}$$

In the model for the case with the crank and linkage mechanism on only one side, clearances in members 2 and 4 are introduced with the following conditions:

$$\begin{aligned}
|\varphi_{2P} - \varphi_2| < \phi_2 &\Rightarrow \varphi_{2P} - \varphi_2 = 0 \\
\varphi_{2P} - \varphi_2 < -\phi_2 &\Rightarrow \varphi_{2P} - \varphi_2 = \varphi_{2P} - \varphi_2 + \phi_2 \\
\varphi_{2P} - \varphi_2 > \phi_2 &\Rightarrow \varphi_{2P} - \varphi_2 = \varphi_{2P} - \varphi_2 - \phi_2 \\
|\varphi_{4P} - \varphi_4| < \phi_4 &\Rightarrow \varphi_{4P} - \varphi_4 = 0 \\
\varphi_{4P} - \varphi_4 < -\phi_4 &\Rightarrow \varphi_{4P} - \varphi_4 = \varphi_{4P} - \varphi_4 + \phi_4 \\
\varphi_{4P} - \varphi_4 > \phi_4 &\Rightarrow \varphi_{4P} - \varphi_4 = \varphi_{4P} - \varphi_4 - \phi_4.
\end{aligned} \tag{8}$$

The case with the crank and linkage mechanism on both sides (Fig. 2) is solved in a similar way, and can be compared with the previous case. The equations of motion for this system are derived using equation (1), where total kinetic energy K, potential energy U and dissipative function R are substituted. Solutions to obtain equations of motion of the system are found, which, after rearrangement for numerical integration, are in the form

$$\begin{aligned}
\frac{d\varphi_{2P}}{dt} &= \omega_{2P} \\
\frac{d\omega_{2P}}{dt}(I_{2P} + I_4\mu_{24}^2) &= -I_4\mu_{24}\nu_{24}\omega_{2P}^2 - C_2(\varphi_{2P} - \varphi_2) + C_4\mu_{24}(\varphi_{4P} - \varphi_4) - k_2(\omega_{2P} - \omega_2) \\
&\quad + k_4\mu_{24}(\omega_{4P} - \omega_4) \\
\frac{d\varphi_{5P}}{dt} &= \omega_{5P} \\
\frac{d\omega_{5P}}{dt}(I_{5P} - I_7\mu_{57}^2) &= -I_7\mu_{57}\nu_{57}\omega_{5P}^2 - C_5(\varphi_{5P} - \varphi_2) + C_7\mu_{57}(\varphi_{4P} - \varphi_7) - k_5(\omega_{5P} - \omega_2) \\
&\quad + k_7\mu_{57}(\omega_{4P} - \omega_7) \\
\frac{d\varphi_{4P}}{dt} &= \omega_{4P} \\
\frac{d\omega_{4P}}{dt} I_{4P} &= -C_4(\varphi_{4P} - \varphi_4) - C_7(\varphi_{4P} - \varphi_7) - M_4 - k_4(\omega_{4P} - \omega_4) - k_7(\omega_{4P} - \omega_7).
\end{aligned} \tag{9}$$

In the equations of motion, transfer functions used are in the form

$$\begin{aligned}
\varphi_4 &= f_{24}(\varphi_{2P}) \\
\dot{\varphi}_4 &= \mu_{24}\dot{\varphi}_{2P} \\
\ddot{\varphi}_4 &= \nu_{24}\dot{\varphi}_{2P}^2 + \mu_{24}\ddot{\varphi}_{2P} \\
\varphi_7 &= f_{57}(\varphi_{5P}) \\
\dot{\varphi}_7 &= \mu_{57}\dot{\varphi}_{5P} \\
\ddot{\varphi}_7 &= \nu_{57}\dot{\varphi}_{5P}^2 + \mu_{57}\ddot{\varphi}_{57}.
\end{aligned} \tag{10}$$

Table 1. Crank and linkage on one side only

	Parameter	Unit	1	2	3	4	5	6	7
			默认值	间隙量	间隙量 打纬力矩	阻尼系数	打纬力矩	扭转刚度 阻尼系数	角速度
	I_{2P}	kg m²	0.005	0.005	0.005	0.005	0.005	0.005	0.005
	I_4	kg m²	0.025	0.025	0.025	0.025	0.025	0.025	0.025
	I_{4P}	kg m²	0.35	0.35	0.35	0.45	0.45	0.35	0.35
扭转刚度	C_2	N m	10^5	10^5	10^5	10^5	10^5	5×10^4	10^5
	C_4	N m	5×10^4	5×10^4	5×10^4	5×10^4	5×10^4	10^4	5×10^4
阻尼系数	k_2	kg m²/sec	7	7	7	7	7	5	7
	k_4	kg m²/sec	29	29	29	33	33	13	29
	ω	sec^{-1}	100	100	100	100	100	100	40
	ϕ_i	rad	0	0.01	0.001	0.001	0.001	0.001	0.001
	M_0	N m	0	0	500	500	0	0	500
杆4的角加速度	$\varepsilon_{4T\max}^{\min}$	sec^{-2}	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−371 466
杆2的角加速度	$\varepsilon_{2P\max}^{\min}$	sec^{-2}	−1666 938	−32 388 54 960	−11 837 13 954	−13 602 15 413	−13 586 15 398	−8501 7945	−3563 5696
杆4P的角加速度	$\varepsilon_{4P\max}^{\min}$	sec^{-2}	−2756 5686	−2848 6975	−2754 7039	−2767 6472	−2814 6033	−5277 4443	−598 1760
杆2的驱动力矩	$M_{2V\max}^{\min}$	N m	−194 193	−426 272	−182 196	−266 232	−271 232	−279 101	−52 50

In the member 4*P*, moment M_4 acts due to the beat-up as given by relation (5), under the conditions given by (6) and (7). The various quantities in equation (10) represent parameters similar to those in the case of a single-end drive. In the case of both-ends drive, clearances are introduced in the members 2, 4, 5 and 7 with the following conditions:

$$|\varphi_{iP}-\varphi_i|<\phi_i \Rightarrow \varphi_{iP}-\varphi_i=0$$

$$\varphi_{iP}-\varphi_i<-\phi_i \Rightarrow \varphi_{iP}-\varphi_i=\varphi_{iP}-\varphi_i+\phi_i$$

$$\varphi_{iP}-\varphi_i>\phi_i \Rightarrow \varphi_{iP}-\varphi_i=\varphi_{iP}-\varphi_i-\phi_i, \tag{11}$$

where $i = 2, 4, 5, 7$.

Table 2. Crank and linkage on both sides

Parameter	Unit	Variant 1	2	3	4	5	6	7
$I_{2P}=I_{5P}$	kg m²	0.005	0.005	0.005	0.005	0.005	0.005	0.005
$I_4=I_7$	kg m²	0.025	0.025	0.025	0.025	0.025	0.025	0.025
I_{4P}	kg m²	0.35	0.35	0.35	0.45	0.45	0.35	0.35
C_2	N m	10^5	10^5	10^5	10^5	10^5	5×10^4	10^5
C_{25}	N m	10^4	10^4	10^4	10^4	10^4	5×10^3	10^4
$C_4=C_7$	N m	5×10^4	5×10^4	5×10^4	5×10^4	5×10^4	10^4	5×10^4
k_2	kg m²/sec	7	7	7	7	7	5	7
k_5	kgm²/sec	2.2	2.2	2.2	2.2	2.2	1.6	2.2
k_4	kg m²/sec	29	29	29	33	33	13	29
k_7	kg m² sec	29	29	29	33	33	13	29
ϕ_i	rad	0	0.01	0.001	0.001	0.001	0.001	0.001
ω	sec^{-1}	100	100	100	100	100	100	40
M_0	N m	0	0	500	500	0	0	500
$\varepsilon_{4T\max}^{\min}$	sec^{-2}	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−2319 2915	−371 466
$\varepsilon_{2P\max}^{\min}$	sec^{-2}	−1292 923	−21 656 24 770	−10 738 10 481	−11 016 10415	−11 074 9674	−8236 7306	−2607 4786
$\varepsilon_{4P\max}^{\min}$	sec^{-2}	−2639 4434	−4664 6822	−3287 5393	−3175 5266	−3203 5257	−4017 4626	−534 1928
$\varepsilon_{5P\max}^{\min}$	sec^{-2}	−2415 1885	−11 493 8034	−3012 3410	−2849 3355	−2848 3375	−4764 3229	−1203 2500
$M_{2V\max}^{\min}$	N m	−201 250	−274 273	−289 327	−281 319	−280 319	−296 167	−52 43

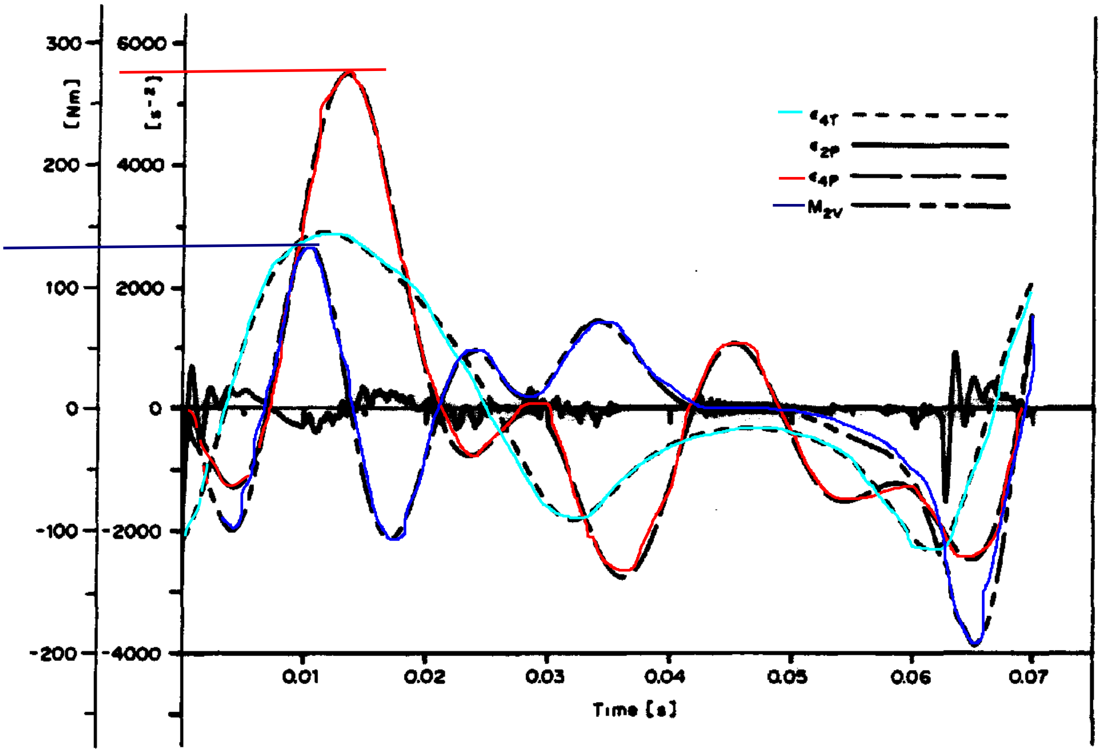

Fig. 4. Angular acceleration and moment of crank and linkage on one side only—variant 1.

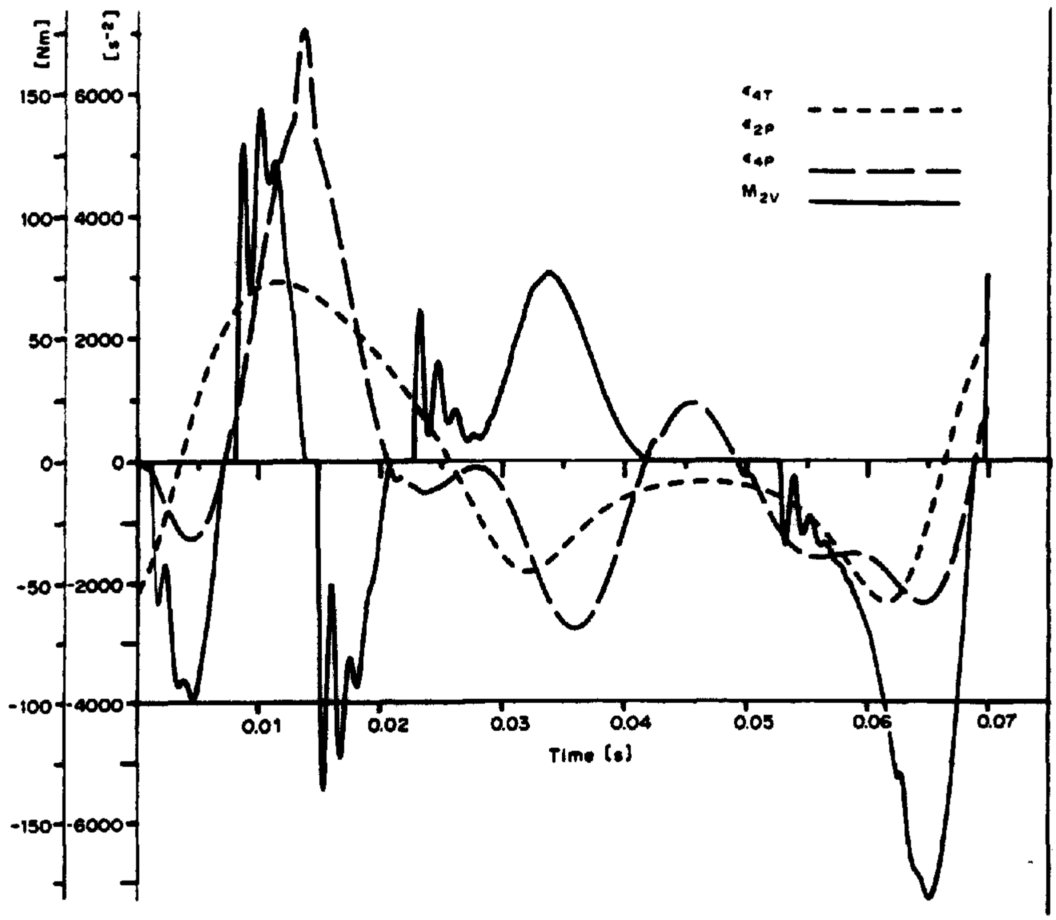

Fig. 5. Angular acceleration and moment of crank and linkage on one side only—variant 3.

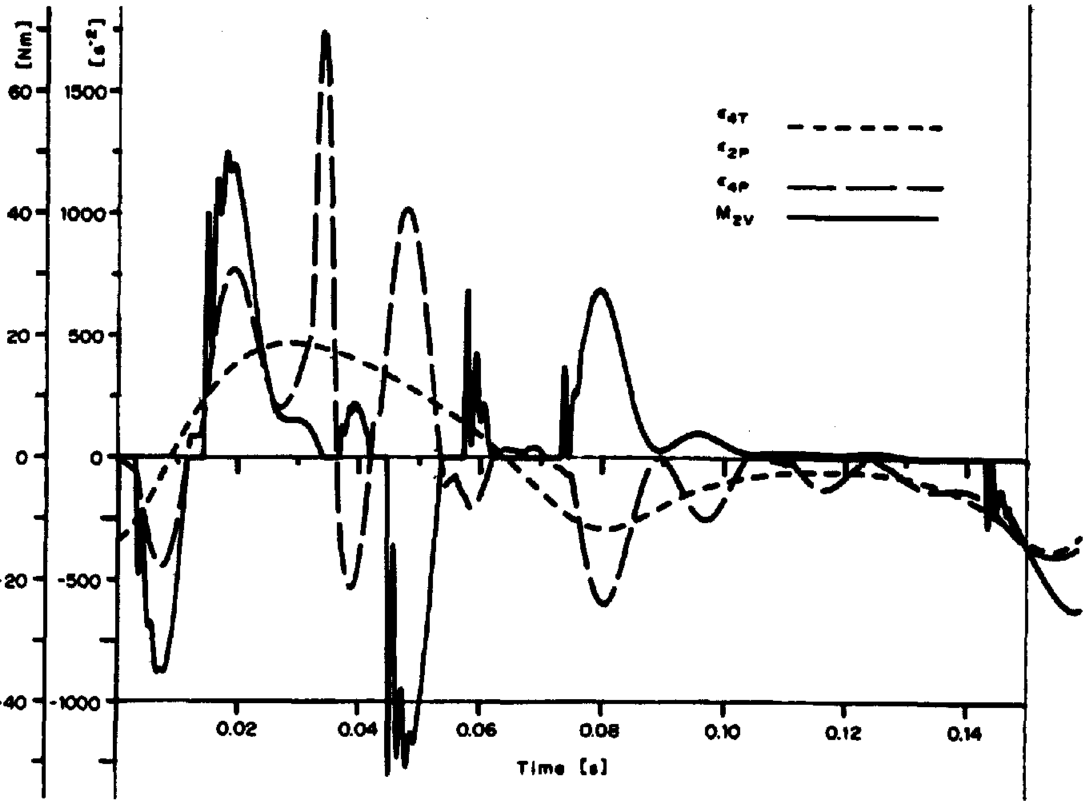

Fig. 6. Angular acceleration and moment of crank and linkage on one side only—variant 7.

The equations of motion for the two cases are integrated by the Runge–Kutta method, using a computer. Input data are determined by experiment or by calculation as far as moment of inertia and torsional rigidity are concerned. Coefficients of damping have been determined using

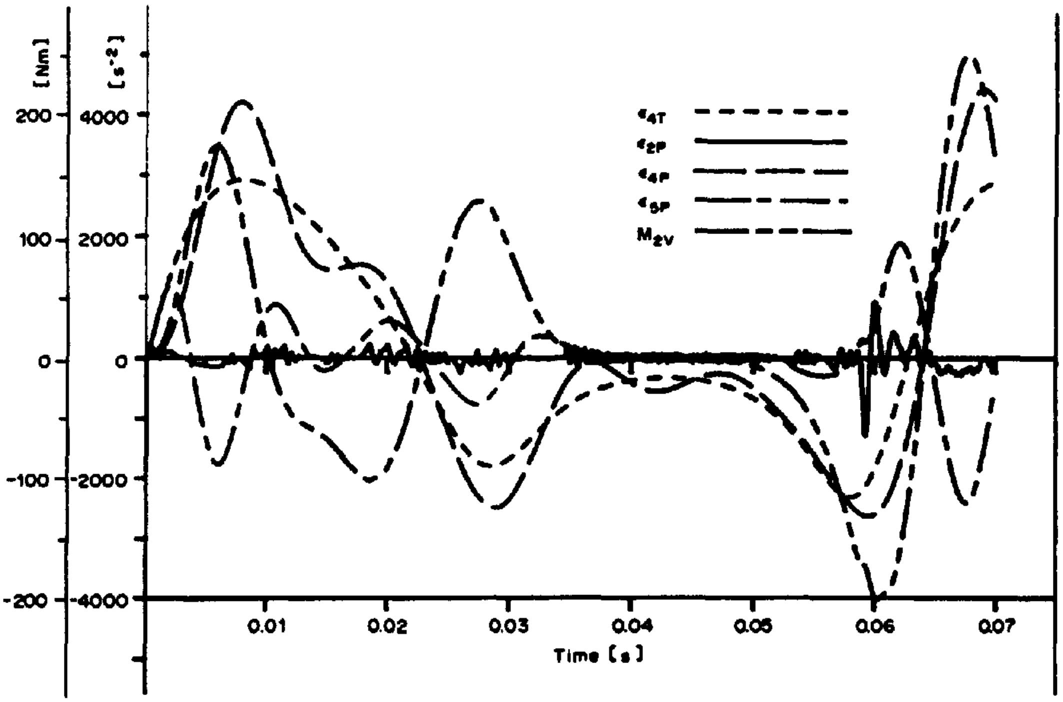

Fig. 7. Angular acceleration and moment of crank and linkage on both sides—variant 1.

logarithmic decrement from the actual measured dependence of a real mechanism and calculated for various mass-related parameters [3].

Integration is started at the position of zero acceleration of the sley, where various members are not loaded and there are no deformations.

这两种情况下的运动方程是由龙格-库塔方法积分，使用计算机。就转动惯量和扭转刚度而言，输入数据由实验或计算确定。阻尼系数已确定使用对数衰减从实际测量依赖的一个真实机构，并计算了各种质量相关参数[3]。
积分开始于筘座零加速度位置，此位置各构件未加载，无变形。

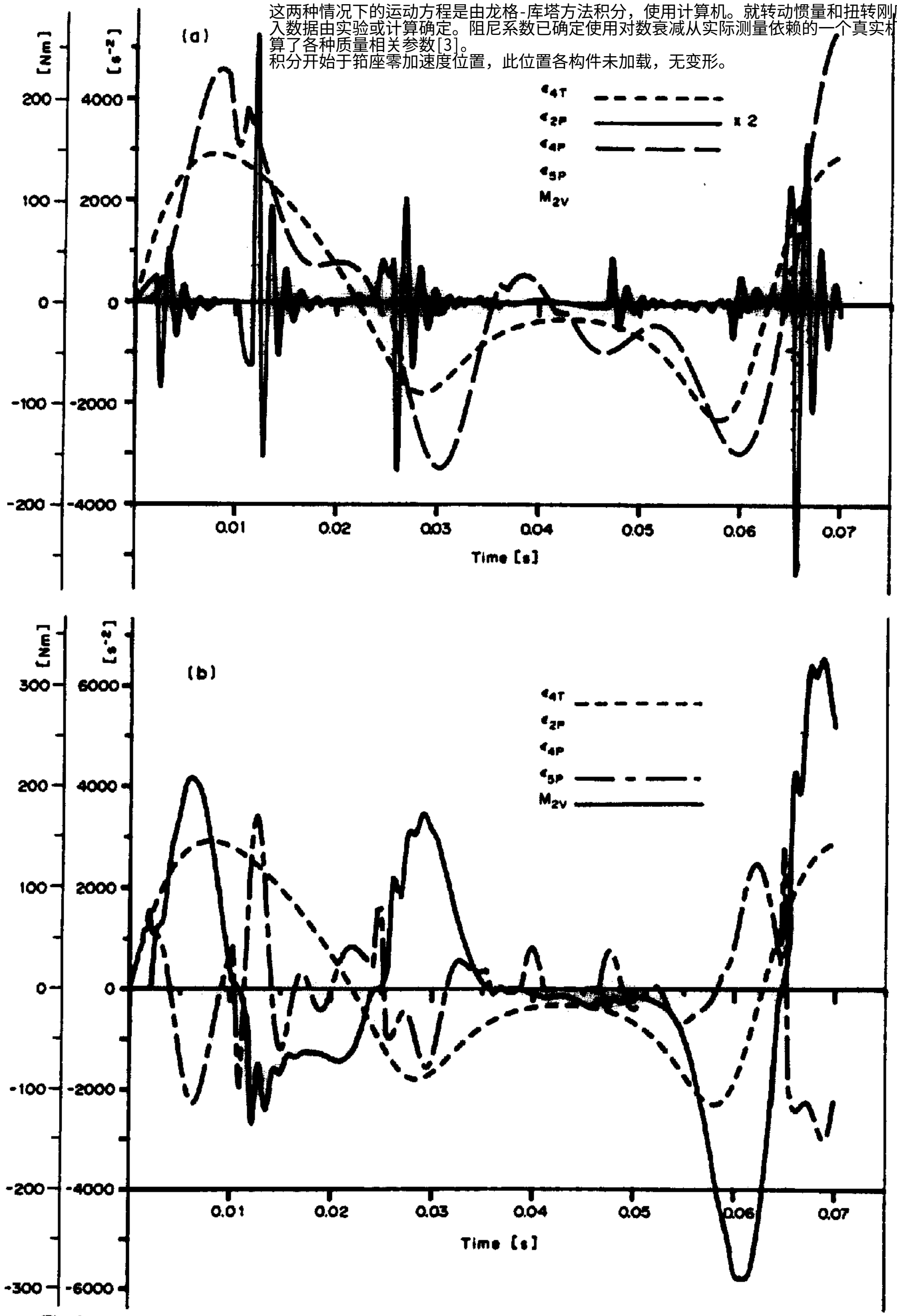

Fig. 8. Angular acceleration and moment of crank and linkage on both sides—variant 3. (Scale of angular acceleration ϵ_{2P} is greater by two times.)

Solutions are found for the same dimensions of corresponding members of the mechanism in both cases.

For Figs 1 and 2 the dimensions have been taken as $l_{2P} = 29$ mm, $l_3 = 37$ mm, $l_4 = 188$ mm and $d = 189$ mm, so that the technological functions of the mechanism, such as the maximum dwell during weft insertion, optimum transformation angle, etc., are ensured to the maximum extent.

In Tables 1 and 2, variant 1 gives the basic input data and the calculated values of various parameters for the two cases considered.

为了说明的目的，7个典型的变种已经选择，在其

DISCUSSION

The basic variant has been solved for an angular velocity of the crank $\omega = 100\ \text{sec}^{-1}$, zero clearance and for zero moment due to beat-up. In the other variants, the effect of change in various parameters on the functioning of the beat-up mechanism has been studied. The characteristics studied are the dependence of angular displacement, velocity and acceleration of individual members, especially of the sley and the driving moment on the crank (link 2 in Figs 1 and 2). The important criteria are the maximum value of these characteristics and the additional frequency of oscillation, which is induced by the non-linearities in the kinematic chain. These are mainly the clearance between the members and the effect of beat-up force.

For the purpose of illustration, seven typical variants have been selected, in which the effect of change in amount of clearance, moment of inertia of the sley, torsional rigidity of the joints between the members, beat-up force and the angular velocity of the crank can be observed.

In the lower portion of Tables 1 and 2, the maximum and minimum acceleration only at the most improtant points of the mechanism and the maximum and minimum values of the driving moment are given.

For illustration, the most important graphical results of acceleration and driving moment, for variants 1, 3 and 7, are given. Figures 4–6 show the results for the case with the crank and linkage mechanism on one side only, and Figs 7–9 show that for the case with crank and linkage mechanism

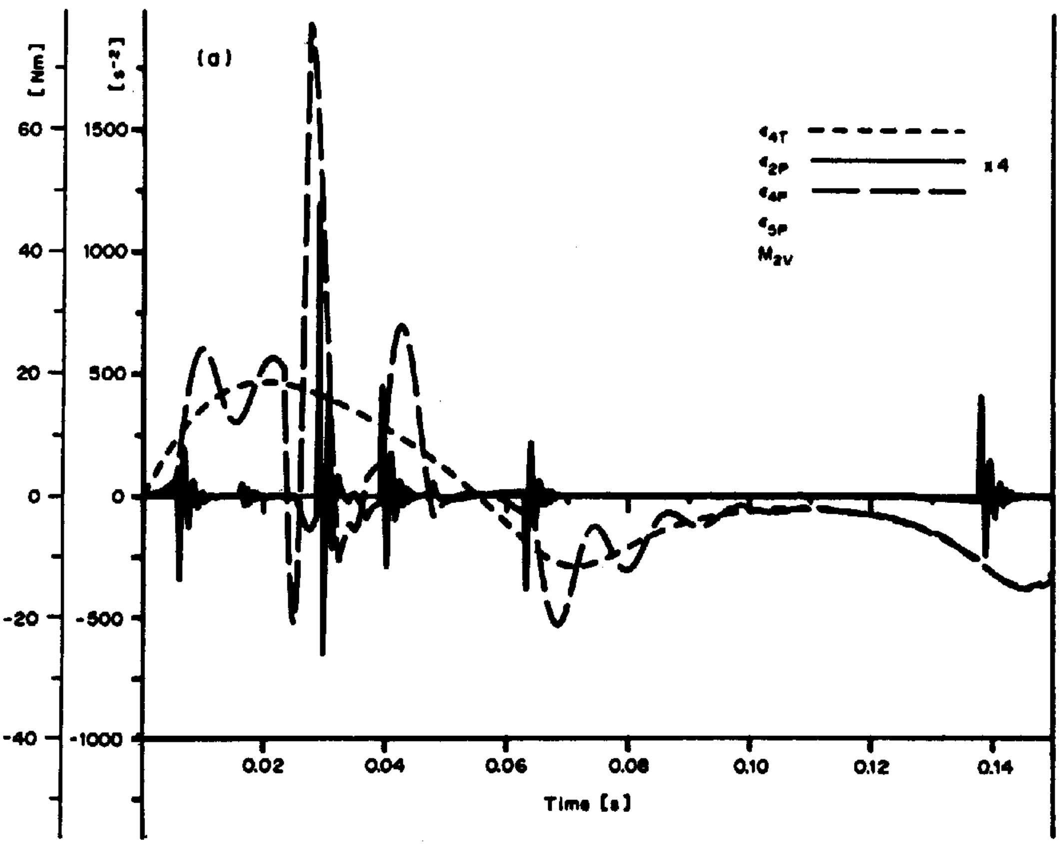

Fig. 9(a)

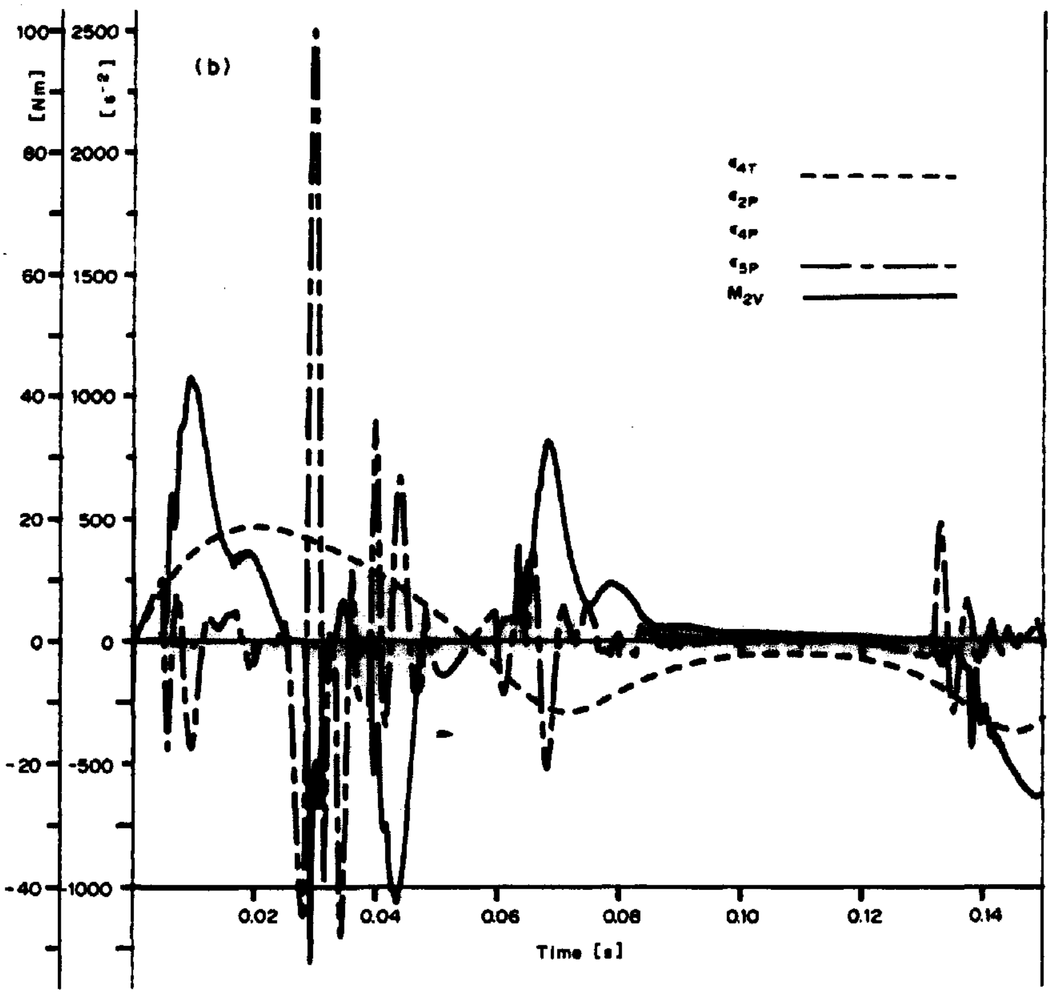

Fig. 9(b)

Fig. 9. Angular acceleration and moment of crank and linkage on both sides—variant 7. (Scale of angular acceleration ϵ_{2P} is greater by four times.)

on both sides. In the figures, curve ϵ_{4T} represents the theoretical values of acceleration of member 4P and curve M_{2V} represents the driving moment on member 2.

For variants 1–5 of the case with one-side drive, the maximum acceleration ϵ_{4P} of member 4P is found to be about twice that of the theoretical maximum ϵ_{4T}. It is interesting to note that lower torsional rigidity (variant 5) leads to significant reduction in the positive side and at the same time to a significant increase in the negative side (the positive maximum approximately corresponds to the beat-up point and the negative side to the weft insertion) of the graphs.

Clearances in the kinematic chain cause significant increase, especially in the driving moment M_{2V} on the crank (variant 2) and a great increase in the acceleration of member 2P. At lower angular velocity, $\omega = 40\,\text{sec}^{-1}$ in variant 7, acceleration at the beat-up increases by four times. This is caused only by the beat-up force M_0, as the maximum driving moment could be expected to be less than that with angular velocity of $\omega = 100\,\text{sec}^{-1}$.

The case with the crank and linkage mechanism on both sides can be analysed in a similar way. A large increase in acceleration of the sley is observed here too. For variant 2, the maximum is twice as great as the theoretical maximum, and for the other variants it is slightly less. It should be noted that beat-up force does not have such a strong effect on the functioning of the mechanism as in the case of single-side drive. The maximum value of driving moment M_{2V} increases insignificantly with respect to increase in clearance. The beat-up force has practically no effect, but acts as a damper for the whole system.

For variants 3 and 4 with clearance, an increase in moment of inertia of the sley 4*P* increases neither the acceleration of the sley nor the driving moment M_{2V}. However, the presence of clearance significantly increases vibration of members 2*P* and 5*P*. These vibrations have small amplitude and, because of smaller mass-related parameters and high rigidity, they have high frequencies.

In Figs 5 and 6, the acceleration curve of member 2*P* is left out for clarity. It is obvious from the graphs that there are undesirable dynamic impacts due to clearances in the kinematic chains. This can be observed in the curve for the driving moment M_{2V}, for variant 3 of the single-side and both-sides drive.

CONCLUSION

本文的目的是建立一个数学模型，可以很容易地模拟所研究的打纬机构的动态特性，相对于基本的非线性驱动力。分析表明了单侧曲柄和连杆的优缺点。加速度曲线表明，采用双侧驱动时筘座的最大加速度比单侧驱动时要小得多。从驱动力矩的角度来看，单侧驱动是有利的，因为由于缺少一组曲柄和连杆，其质量会有差异。

The aim of this paper has been to formulate a mathematical model which can easily simulate the dynamic characteristics of the studied beat-up mechanisms, with respect to the basic non-linear driving force. The analysis indicates the advantages and disadvantages of the two cases studied—crank and linkage on one side only or on both sides. The acceleration curves shown that maximum acceleration of the sley is much smaller in the case of both-sides drive than that of single-side drive. From the point of view of the driving moment, single-side drive is advantageous, because of the difference in the mass as a result of the absence of one set of crank and linkages.

双侧曲柄和连杆机构结构紧凑，同时对运动链的间隙(比单侧驱动多)更敏感。该系统作为一个整体，在制造精度和保持公差方面要求更严格，在任何情况下，它对加载的敏感性较低，因为布掉落的打击阻力。

The mechanism with crank and linkage on both sides is more compact and at the same time more sensitive to the clearances in kinematic chains (which are more in number than in single-side drive). The system as a whole is more rigid and more demanding in terms of precision in manufacture and maintaining tolerances. In any case, it is less sensitive to loading as a result of the beat-up resistance from the fell of the cloth.

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THEORETISCHE ANALYZE DER DYNAMIK 4-GLIEDRIGEN MECHANISMUS DER WEBMASCHIENE

Zusammenfassung—Beim Entwurf des Anschlagmechanismus der Webmaschine treten Probleme einer objektiven Bewertung dessen einzelner Parameter auf. Es handelt sich vor allem um den Einfluß der Grösse des Eintrags—und des Ladenschwingungswinkels mit Webeblatt, der Struktur der Mechanismen, der Konstruktions-dispositien u.ä. auf die einwandfreie Funktion und ruhigen Lauf des ganzen Mechanismus. Im Beitrag wird die geeignete Methode gezeigt, mit der die Überprüfung dynamischer Eingenschaften der entworfenen Struktur leicht und schnell durchzuführen ist. Sie basiert auf der Aufstellung des mathematischen Models eines Realmechanismus mit Hilfe von Lagrange-Gleichung der zweiten Art. Die Bewegungsgleichungen berücksichtigen die Massenparameter der Mechanismus, Bindungen unter seinen Gliedern, technologische Belastungen sowie die Nichtlinearitäten, die ist System durch das Konstruktionsspiel und die Übersetzung des benutzten Mechanismus eingetragen werden.