Dynamic Analysis of the Press Automation

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***Abstract*—In this paper, we present a geometric exploitation of the d'Alembert–Lagrange equation (or alternatively, Lagrange form of the d'Alembert's principle) on a Riemannian manifold. We develop the d'Alembert–Lagrange equation in a geometric form, as well as an explicit analytic form with respect to an arbitrary frame in a coordinate neighborhood on the configuration manifold. We provide a procedure to determine the governing dynamic equations of motion. Examples are given to illustrate the new formulation of dynamic equations and their relations to alternative ones. The objective is to provide a generalized perspective of governing equations of motion and its suitability for studying complex dynamic systems subject to nonholonomic constraints.**

***Index Terms*—press automation, d'Alembert–Lagrange equation, computed plot, numerical algorithm**

I Introduction

Our objective in what follows is to present a formulation of the d'Alembert–Lagrange equation on a configuration manifold. We regard this as a geometric version of a variational approach involving quasi-coordinates. To facilitate the derivation we cast a dynamic system onto a Riemannian manifold [1]. This geometric setup provides smooth and convenient coordinate transformations and it allows the dynamic system to be studied using the powerful tenets of Riemannian geometry [2]. Specifically, we derive a geometric (i.e., coordinate free) formulation of the d'Alembert–Lagrange equation. We demonstrate that in a coordinate neighborhood on the configuration manifold, an explicit form of the d'Alembert–Lagrange equation can be obtained with respect to an arbitrary frame on the tangent bundle [3]. We further show that if there exists a set of independent 1-forms which constitute a basis of the dual frame, then the independent equations of motion can be determined. This new approach permits the deployment of quantities which best describe the configuration, constraints or motion of a subject system, and it yields motion equations in concise form as with Kane's equations and Boltzmann–Hamel equations [4]. To illustrate the application and effectiveness of the new formulation, a single rigid body, a general nonholonomic system, and a rolling disk are considered. The governing equations of motion for such systems are developed and are equivalent to those obtained by other methods. The more complicated the device of the lever transmission, analysis and the synthesis of such mechanisms will be difficult. Therefore, a two-part structural group will be used for practice [5], [6]. Lever transmission consists of the main mechanism consisting of four parts of the structural group, their application in technology will not be [7], [8]. The reason is the difficulty of spreading the practice of design engineers in the study of the mechanism of the structural group of the four parts. Automation and automatic press is carried out by means of processing of two lever mechanism of structural group functionality [9]. In this regard, based on the structural group of the four parts, mechanisms and planning machines, the synthesis of dynamic planning, the processing of a numerical algorithm is a versatile issue, consisting of a structural group of four parts.

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II Materials and Methods

Dynamical equations on the first degree of freedom of the mechanism can be calculated from the basis of the algorithmic system without resorting to the cumbersome approach of solving the reduced force and mass [1]. The approach excludes the process of differentiating cumbersome expressions and allows us to construct an explicit dynamic equation with fairly good approximations [2]. The executive mechanism of the exhaust press machine is a flat crank-and-rod mechanism of the fourth class, represented in Fig. 1.

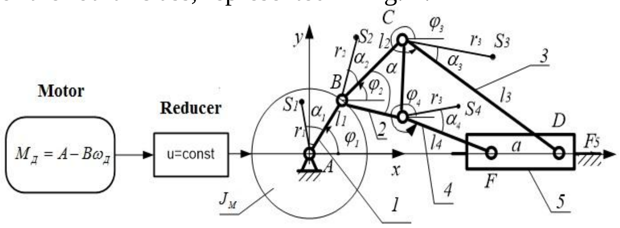

Figure 1. Kinematic model of six-bar mechanical press

In a variant of the dynamic equation on the first degree of the mechanism is presented:

$$J(q)\ddot{q} + 0{,}5J'(q)\dot{q}^2 = Q_D - Q_C(q, \dot{q}, t) \qquad (1)$$

where, q - accumulated coordinates, $J(q)$, the reduced moment of inertia and its production from the accumulated coordinates, Q_D , Q_C The forces accumulated in motion and accordingly resistance forces

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[3]. In this algorithm (1) the d'Alembert-Lagrange equations are applied:

$$Q_D + \sum_{i=1}^{N}\left[\left(\vec{P}_{Ci} + \vec{F}_i\right)\frac{\partial \vec{r}_{Oi}}{\partial q} + \left(\vec{M}_{Oi}^{P_C} + \vec{M}_{Oi}^{F}\right)\frac{\partial \vec{\varphi}}{\partial q}\right] = 0 \quad (2)$$

On the basis of Eq. (2) from Eq. (1), by means of variation, it is possible to obtain discrete values of the inertia moment and its derivative, as well as the reduced moment of the resistance force mechanism [4]. The digital calculated algorithm of the mechanism includes the following main values: the equations of motion of a strictly secured press machine are indicated below:

$$\begin{aligned} &J_n\ddot{\varphi}_1 + \frac{1}{2}\frac{dJ_n}{d\varphi_1}\dot{\varphi}_1^2 = Q = M_{\ddot{A}} + M_C(\varphi_1, \dot{\varphi}_1) \\ &\tau \cdot M_{\ddot{A}} + M_{\ddot{A}} = A - B\dot{\varphi}_1 \end{aligned} \quad (3)$$

$\tau \cong 0$, $M_{\ddot{A}} = A - B\dot{\varphi}_1$ - static motor characteristics.

III Methods of Dynamic Synthesis of a Press Machine with a Crank Mechanism of the Fourth Class

On the basis of Eq. (3) or Eq. (1), the functional created as a complete function:

$$\Re = \sum_{i=1}^{N}\left[J_{\Pi,i}\ddot{\varphi}_{1,i} + \frac{1}{2}\left(\frac{dJ_{\Pi}}{d\varphi_1}\right)_i \dot{\varphi}_{1,i}^2 - Q_i\right]^2 \quad (4)$$

Given the static characteristics of the engine Eq. (3), the required parameters of the machine aggregate are determined from the condition of a minimum of the root-mean-square value of the sum of the moments of the driving forces, the resistance forces and the inertia forces Eq. (4), reduced to the crank, for n positions of the mechanism [5].

The input parameters of the synthesis of the machine aggregate are the strength of the technological resistance $F_5 = F_Д$ the angle of the crank φ_1, the geometrical dimensions of the links of the mechanism l_1, l_2, l_2', l_3, l_4, a, α, the gear ratio of the gear train u, and the crank motion law l_1. Angular velocity $\omega_1 = \dot{\varphi}_1 = \omega_{CP} + 0.5\delta \cdot \omega_{CP}\cos(\varphi_1 + \nu_1)$ and angular acceleration of the driven crank $\varepsilon_1 = \ddot{\varphi}_1 = -0.5\delta \cdot \omega_{CP}\sin(\varphi_1 + \nu_1)\cdot\dot{\varphi}_1$. The output parameters of the synthesis will be the inertial parameters, the mass and moment of inertia with respect to the centering mass and the coordinates with respect to the centering mass of the crank S_k. These parameters are presented in this form m_k, J_{S_k}, x_{S_k}, y_{S_k}, here k the link number [6].

The equations of the inertial moment of the aggregate will be as follows:

$$I_{np} = I_{\partial в}\cdot\mu^2 + I_1 + I_M + \sum_{k=2}^{4} y_k\varphi_k'^2 + \sum_{k=2}^{5} m_k\left(x_{S_k}'^2 + y_{S_k}'^2\right) \quad (5)$$

The remaining parameters are determined by the following equation:

$$\frac{\partial I_{np}}{\partial \varphi_1} = 2\sum_{k=2}^{4} I_k\varphi_k'\varphi_k'' + 2\sum_{k=2}^{5} m_k\left(x_{S_k}'x_{S_k}'' + y_{S_k}'y_{S_k}''\right) \quad (6)$$

$$Q = M_{\partial в}\cdot\mu + F_5x_{S_5}' + m_1g(b_1\sin\varphi_1 - a_1\cos\varphi_1) - m_2gy_{S_2}' - m_3gy_{S_3}' - m_4gy_{S_4}' \quad (7)$$

The value $M_{\partial в}$ is indicated by the static parameters of the induction motor:

$$M_{\partial в} = A_1 - B_1\mu\dot{\varphi}_1 \quad (8)$$

In formulas Eq. (5-7), the velocity, acceleration and center of gravity of the link are calculated. The speed to be considered and the anologue are discussed above [7]. For example, in formulas Eq. (1) and Eq. (2), the speed and the center of gravity of the acceleration are determined by two links [8]. But we will validate the values of the equation [9]. In mechanics for the basis we take only the mass of the crank and link, that is $m_2 = m_3 = m_4 = 0$.

then:

$$\begin{cases} I_{np} = I_0 + m_5x_{S5}'^2, \qquad I_0 = I_{\partial в}\cdot\mu^2 + I_1 + I_M, \\ \dfrac{\partial I_{np}}{\partial \varphi_1} = 2m_5x_{S5}'x_{S5}'', \\ Q = (A_1 - B_1\mu\dot{\varphi}_1)\mu + F_5x_{S5}' + m_1g(b_1\sin\varphi_1 - a_1\cos\dot{\varphi}_1). \end{cases} \quad (9)$$

We put these values in the formula Eq. (4) and obtain the function of deviation Δ:

$$\Delta = I_0\ddot{\varphi}_1 + m_5x_{S_5}'^2\ddot{\varphi}_1 + m_5x_{S_5}'x_{S_5}''\dot{\varphi}_1^2 - A_1\mu + B_1\mu^2\dot{\varphi}_1 - F_5x_{S_5}' + m_1ga_1\cos\varphi_1 - m_1gb_1\sin\varphi_1 \quad (10)$$

The deviation function Δ can be considered in the form of a polynomial and the last criterion will be in the following form [10]:

$$\Re = \sum_{i=1}^{N}\left[\sum_{j=1}^{5} P_j f_j(\varphi_{1i}) - F(\varphi_{1i})\right]^2 \quad (11)$$

In formulas Eq. (10) and Eq. (11), the following values:

$$\begin{aligned} &x_{S5} = x_F + a/2;\ x'_{S5} = -l_1 \sin\varphi_1 - l_2 \sin\varphi_2\varphi'_2 - l_3 \sin\varphi_3\varphi'_3, \\ &x''_{S5} = -l_1 \cos\varphi_1 - l_2 \cos\varphi_2\varphi'^2_2 - l_2 \sin\varphi_2 \cdot \varphi''_2 - l_3 \cos\varphi_3 \cdot \varphi'^2_3 - l_3 \sin\varphi_3 \cdot \varphi''_3. \end{aligned} \tag{12}$$

$$\begin{cases} P_1 = m_1 a_1; \quad P_2 = m_1 b_1; \quad P_3 = I_0; \quad P_4 = A_1; \quad P_5 = B_1; \quad P_6 = m_5, \\ f_1(\varphi_1) = g\cos\varphi_1; \quad f_2(\varphi_1) = -g\sin\varphi_1; \quad f_3(\varphi_1) = \ddot{\varphi}_1, \\ f_4(\varphi_1) = -\mu; \quad f_5(\varphi_1) = \mu^2\dot{\varphi}_1; \quad f_6(\varphi_1) = x'^2_{S5}\ddot{\varphi}_1 + x'_{S5}x''_{S5}\dot{\varphi}^2_1, \\ F(\varphi) = F_5 x'_{S5}. \end{cases} \tag{13}$$

The minimum condition for the function Eq. (11) gives the system of equations which determines the coefficients P1, P2, ..., P6:

$$\frac{\partial R}{\partial P_k} = 0,\ \ k = 1,2,\ldots.6 \tag{14}$$

This system will be in the following form:

$$\sum_{i=1}^{6} c_{ij} P_j = r_i,\ \ k = 1,2,\ldots.6 \tag{15}$$

here:

$$c_{ij} = c_{ji} = \sum_{k=1}^{N} f_i(\varphi_{1k}) f_j(\varphi_{1k}) \tag{16}$$

$$r_i = \sum_{k=1}^{N} F(\varphi_{1i}) f_i(\varphi_{1k}),\ i = j = 1,2,\ldots.6$$

After determining P1, P2, ..., P6, the physical parameters are determined [11]. If we accept $m_2 \neq 0$, then the values will be nonlinear, that is $P_i = P_i(m_2, I_2, a_2, b_2)$ and $i = 1,2,\ldots.6$ it is added to the functional minimization by formula $R = R(m_2, I_2, a_2, b_2)$ in Eq. (11).

The solution to the equations indicated above depends on the following equation:

1. $\varphi_1 \in [0, 2\pi]$ n divide into equal values;
2. If the average angular velocity of the crank ω_{cp} is given and the coefficient of the non-equilibrium δ, then for the angular velocity we can take the first values $\omega_1 = \dot{\varphi}_1 = \omega_{cp} + 0.5\delta\omega_{cp}\cos(\varphi_1 + \nu_1)$;
3. Angular acceleration of the crank $\varepsilon_1 = \ddot{\varphi}_1 = -0.5\delta\omega_{cp}\sin(\varphi_1 + \nu_1)\cdot\dot{\varphi}_1$;
4. From the formula Eq. (16) we calculate the coefficients c_{ij} and r_i;
5. In Eq. (15) we solve the linear system and determine the desired parameter P_k;
6. From the formula Eq. (14) we determine the last physical parameters.

IV Results and Discussion

According to the scheme of 3D MD Adams, you can see the internal structure of the integrated planned a new type of automatic press hammer-stamping mechanism (see Fig. 2), considering the schema in the process links the main role of parts, we see that each affects a different power, thus loses its working strength. And the expectation based on the way analysts and diagrams kinematics we do not consider the effect to the mechanism of links, however, the value of which we do not consider has a significant influence mechanism, so that all values were taken into account to come to the aid of MD Adams (see Fig. 3-11).

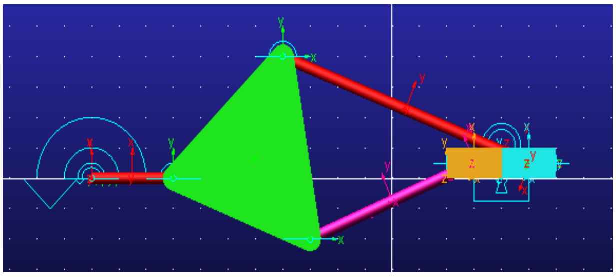

Figure 2. Six bar linkage motion simulation in MD Adams

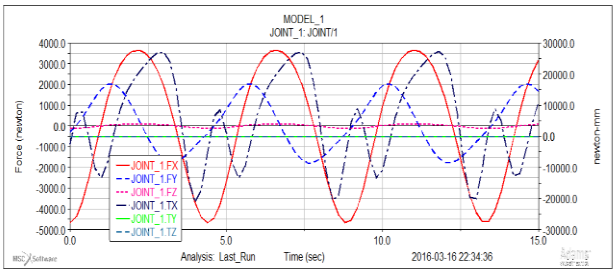

Figure 3. Computed plot of the force and momentum joint 1

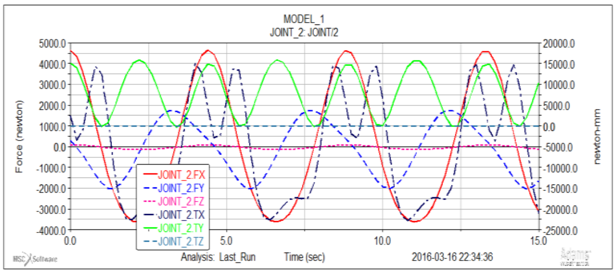

Figure 4. Computed plot of the force and momentum joint 2

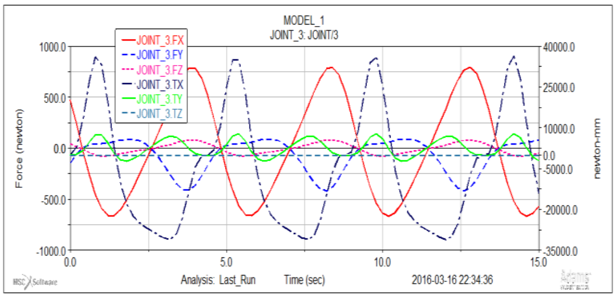

Figure 5. Computed plot of the force and momentum joint 3

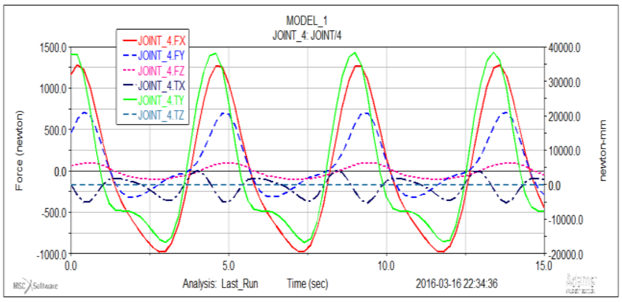

Figure 6. Computed plot of the force and momentum joint 4

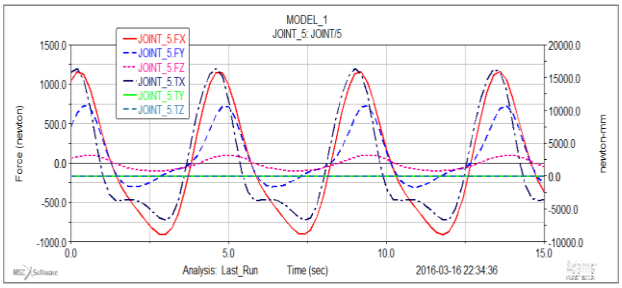

Figure 7. Computed plot of the force and momentum joint 5

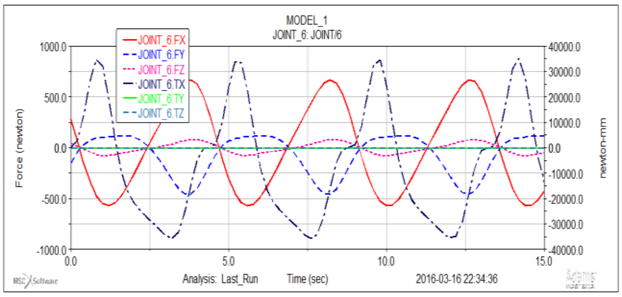

Figure 8. Computed plot of the force and momentum joint 6

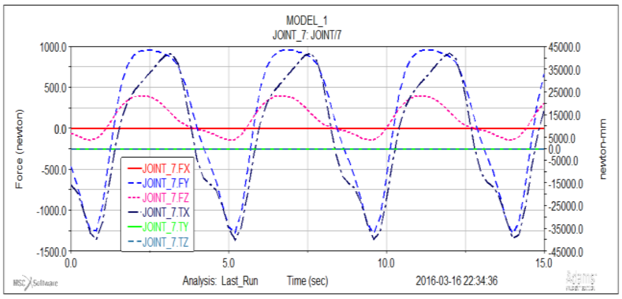

Figure 9. Computed plot of the force and momentum joint 7

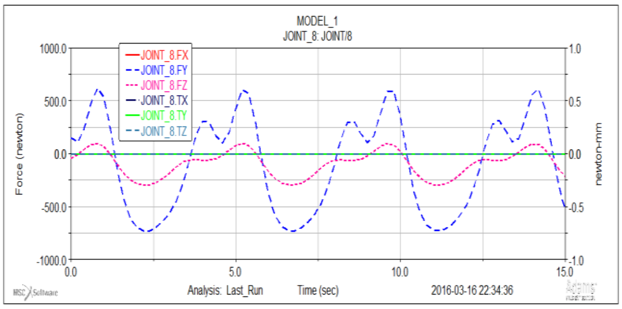

Figure 10. Computed plot of the force and momentum joint 8

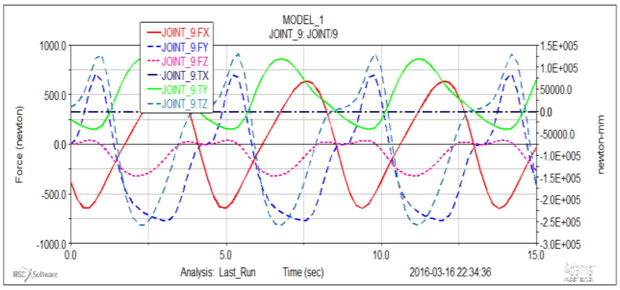

Figure 11. Computed plot of the force and momentum joint 9

V Conclusions

In the current study, the dynamic analysis of a six-bar linkage of a mechanical press for deep drawing has been investigated developing a MD Adams program. An analytical method has been used for the determination of the displacement, velocity and acceleration of the links and the simulation of mechanism motion. The force analysis considering the joint friction is performed with iterative procedure, applying the D'Alembert principle. It is observed that due to the low values of the friction forces and moments, the solution converges after one iteration step. The developed program can be useful for the optimization of the press design considering different constraints.

References

[1] M. Lesser, "A geometrical interpretation of Kane's equations," *Mathematical and Physical Sciences*, vol. 436, pp. 69–87, 1992.

[2] X. Liu, "A lie group formulation of Kane's equations for multibody systems," *Multibody Syst. Dyn*, vol. 20, pp. 29–49, 2008.

[3] A. Zhauyt, I. Zhahanova, G. Smailova, U. Murzakhmetova, L. Kurmangalieva, A. Kurenbay, "The kinematic analysis of the third class mechanism," *Vibroengineering Procedia*, vol. 12, pp. 208-212, 2017.

[4] A. Zhauyt, K. Alipov, A. Sakenova, A. Zhankeldi, R. Abdirova, Z. Abilkaiyr, "The synthesis of four-bar mechanism," *Vibroengineering Procedia*, vol.10, pp. 486-491, 2016.

[5] A. Zhauyt, "The substantiating of the dynamic parameters of the shaking conveyor mechanism," *Vibroengineering Procedia*, vol. 5, pp. 15-20, 2015.

[6] E. Askarov, A. Zhauyt, Z. Abilkaiyr, A. Zhankeldi, B. Naurushev, "A new type cam-screw mechanical press," *Engineering for Rural Development*, vol. 16, pp. 36-41, 2017.

[7] A. Zhauyt, G. Mamatova, K. Alipov, A. Sakenova, R. Abdirova, "The kinematic analysis of flat lever mechanisms with application of vector calculation," *Vibroengineering Procedia*, vol. 8, pp. 1-5, 2016.

[8] S. Rodrigo, J. Jamisola, M. Carlos, and I. Frank, "Modular relative Jacobian for combined 3-Arm parallel manipulators,"

Inter. Jour. Mech. Eng. and Robot. Res., vol. 5, no. 2, pp. 90-95, 2016.

[9] C. C. Cheng and C. Tsai, "Visual servo control for balancing a Ball-Plate system," *Inter. Jour. Mech. Eng. and Robot. Res.*, vol. 5, No. 1, pp. 28-32, 2016.

[10] A. Zhauyt, G. Mamatova, G. Abdugalieva, K. Alipov, A. Sakenova, A. Alimbetov, "The kinematic analysis of flat leverage mechanism of the third class," *IOP Conference Series: Materials Science and Engineering*, vol. 250, pp. 1-6, 2017.

[11] J. Musayev, A. Zhauyt, "Analysis of disturbing influence of traffic load on soil body," *Advances in Materials Science and Engineering*, vol. 2015, pp. 1-7, 2015.

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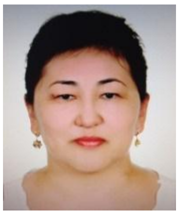

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