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On modernization of the lever mechanism in a semi-dry pressing machine

N N Dubinin, S A Mikhailichenko and S I Goncharov

Belgorod State Technological University named after V.G. Shoukhov, 46, Kostyukova st., Belgorod, 308012, Russia

E-mail: nndubinin@mail.ru

Abstract. The paper presents the criteria for the modernization of the lever mechanism of the pressing machine, developed on the basis of the analysis of semi-dry pressing of silicate bricks. The constructive restrictions of the movement of the mechanism links are considered. This allows one to select new layout solutions based on the obstacles within the working area of the prototype. A mathematical model of the silicate brick pressing system is developed. It is shown how to change the initial data in the model to form an array of link lengths of the new mechanism. A method for determining the domain of rational solutions within this array is proposed. The use of these sizes in the modernization of the press will improve the quality of products.

1. Introduction

The increase in the efficiency of semi-dry pressing of building bricks is one of the conditions for innovative transformation of the industry of building materials. For many enterprises, the economically beneficial practical realization of these processes is bound with the implementation of currently working mechanical equipment [1, 2]. The research of the pressing process [3, 4] and development engineering of pressing equipment upgrade [5] have demonstrated a promising perspective of this direction.

Its current development state [6–8] allows concluding that the task of the pressing equipment upgrade involves both provision of maximum possible intensity of pressing process, and concurrent minimization of the metal consumption by structural transformations and labor intensity of maintenance operations.

In this connection, this multi-parameter problem requires a thorough investigation of all the solutions by mathematical modeling of the press mechanical system movements and estimation of their effect on the quality of the products.

2. Kinematic synthesis of the semi-dry pressing lever mechanism

A press for silica brick pressing is composed of several mechanisms connected to each other by members that are put into motion from a single motor [1, 3]. Since the motor has really rigid mechanical characteristic, the angle velocity of its shaft is nearly constant. The press mechanisms are synchronized by a path, so the time for each operation and period of the cycle are constant.



For example, the operation of brick pressing is performed at crank angle φ_1 (figure1) from position $\varphi_1=\varphi_{1\text{beg}}$ to position $\varphi_1=\varphi_{1\text{end}}$. Pressing piston 8 of the pressing mechanism reaches its maximum upper position at $\varphi_1=\varphi_{1m}$ and begins reciprocal idle run, during which in position $\varphi_1=\varphi_{1\text{end}}$ it disengages with the turnplate.

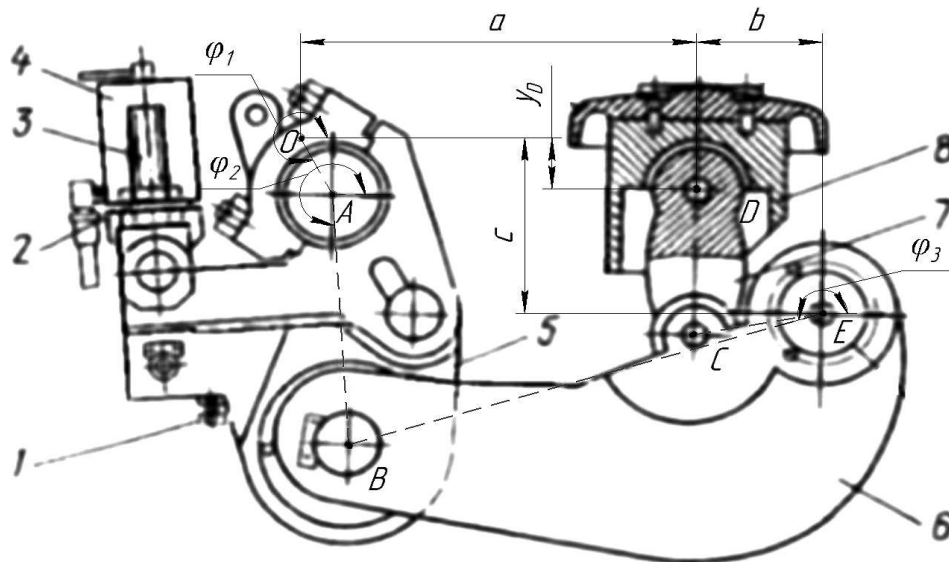


Figure 1. The kinematic scheme of the pressing lever mechanism: 1) adjusting screw; 2) and 3) parts of auxiliary threaded joint; 4) hydrocylinder of pressure stabilizer; 5) split rod AB; 6) lever BE; 7) link CD; 8) pressing piston; φ_1 , φ_2 and φ_3 are turning angles of crank OA, members AB and BE, correspondingly; a , b and c are the coordinates of kinematic pairs connected with the stand.

The efficiency of construction brick pressing is mainly connected with the pressing piston motion. It depends on the scheme of the lever mechanism and size of its members. Taking into account the requirements of minimum modernization of the press design, let us leave the scheme of its pressing mechanism unchanged. Let us only consider the change in the lengths L_1 , L_2 , L_3 and L_4 for crank, rod, lever and link, correspondingly. In addition, let us leave unchanged dimensions a , b and c that determine the position of the crankshaft (point O), axis of the pressing lever (point E) and symmetry axis for turnplate molds coinciding at its stops with the guide of piston 8. Let us assume that the design and position of the turnplate remain unchanged.

Let us find the transfer function of the pressing machine [1, 3, 9]. The function of the position of lever 6 and piston 5 is as follows:

$$\varphi_3 = \arccos((L_3^2 + s^2 - L_2^2)/(2 \cdot L_3 \cdot s)) + \arctg((-c + L_1 \cdot \sin(\varphi_1))/((a+b) - L_1 \cdot \cos(\varphi_1))) \quad (1)$$

where

$$s = (-c + L_1 \cdot \sin(\varphi_1))/\sin(\varphi_s), \quad (2)$$

$$\varphi_s = \arctg((-c + L_1 \cdot \sin(\varphi_1))/((a+b) - L_1 \cdot \cos(\varphi_1))) \quad (3)$$

$$y_D = L_c \cdot \sin(\varphi_3) + L_4 \cdot \sin(\varphi_4) - c \quad (4)$$

here L_c is the coordinate on the link joint lever, φ_4 is link turning angle:

$$\varphi_4 = \arccos(-(b + L_c \cdot \cos(\varphi_3))/L_4) \quad (5)$$

When organizing the calculation of function (4) over the motion cycle $\varphi_1=0...2\pi$ in a definite range of member size alteration, let us consider only those solutions that do not change the connection in terms of the position between the pressing piston 8 and the turnplate with molds.

To do so, let us first calculate functions (1)...(5) for the dimensions of the prototype of the pressing mechanism (i.e. before the modernization). From the calculation, we determine angle φ_{1m} in the maximum upper position of the slide $y_{D\text{max}}=y_D(\varphi_{1m})$ and the distance between the lever joint axis and the turnplate lower surface:

$$y_{zn} = y_{Dmax} - S_{prs} \quad (6)$$

where S_{prs} is the pressing travel of the prototype. Then, for the synthesized mechanism at y_{Dmax}^* the maximum piston travel is:

$$S_{nw} = y_{Dmax}^* - S_{ppr} \quad (7)$$

where S_{ppr} is the decrease in the slide travel after prepressing. The pressing travel is:

$$S_{prs_nw} = S_{nw} - y_{zn} \quad (8)$$

Then, we determine the range of the turning angles of the crank $\varphi_{1beg} \dots \varphi_{1end}$, at which piston 8 of the prototype engages with the turnplate. By using array $y_D(\varphi_1)$ at $\varphi_1 = 0 \dots 2\pi$, this event is identified if the following inequality is false:

$$y_{zn} - y_D(\varphi_1) \geq 0 \quad (9)$$

In such conditions, the pressing phase corresponds to the range of crank turning angles $\varphi_{1beg} \dots \varphi_{1m}$.

After determining for the synthesized mechanism range $\varphi_{1beg}^* \dots \varphi_{1end}^*$ under the following conditions:

$$\varphi_{1beg}^* < \varphi_{1end}^*, \varphi_{1beg}^* \geq \varphi_{1beg}, \varphi_{1end}^* \leq \varphi_{1end} \quad (10)$$

Let us leave the mechanism for further investigation. It involves the check of Grashoff's rule validity:

$$L_1 < L_2 < L_3 < \sqrt{(a+b)^2 + c^2} \quad (11)$$

$$L_1 + \sqrt{(a+b)^2 + c^2} \leq L_2 + L_3 \quad (12)$$

Besides, for all synthesized mechanisms, the pressing travel—or, considering (6), (7), and (8), the maximum upper position of the slide—should not be drastically changed. Then:

$$|y_{Dmax}^* - y_{Dmax}| \leq \Delta \quad (13)$$

Reasonably specified error Δ , in the majority of cases, would not ensure the satisfaction of this condition. Thus, for the mechanism synthesis, it is realized through the determination of the link length L_4 by the maximum upper position (at $\varphi_1 = \varphi_{1m}^*$) of pressing lever 6:

$$L_4 = \sqrt{\left(L_c \cdot \cos(\varphi_3 - \psi) \right)^2 + \left(S_{nw} - L_c \cdot \sin(\varphi_3 - \psi) \right)^2}$$

where ψ is the angle between lines EC and EB (Fig. 1). In this case, the check for validity using condition

$$y_{Dmax}^*(\varphi_{1end}^*) < y_{zn} \quad (14)$$

can be provided by the adequate correction of error Δ .

By cyclic calculations in *Matlab (MathWorks, Inc.)* with variable L_1 , L_2 and L_3 in functions (1)...(5) considering boundaries (6)...(13), by calculating L_4 we determine: a) the number of mechanisms n , i.e. combinations of dimensions L_1 , L_2 and L_3 that passed the check under modernization condition; b) regularity of piston travel $S^* = y_D^*(\varphi_1)$ at the pressing phase at $\varphi_1 = \varphi_{1beg} \dots \varphi_{1m}$.

To process these results for determining rational solution, we should quantitatively estimate the properties of function S^* .

3. Identification of rational solutions

The studies of silica brick pressing [4, 6] have established that at constant pressing time, the positive effects can be achieved through increased brick strength. To do so, the pressing force first should most intensely increase, and then gradually approach the maximum value in the end of the travel. The increased duration of this process also has positive effect on the product quality.

The pressing force, under the conditions of uniform distribution along the press-tool surface, is proportional to the pressure that can be estimated using the equation of P.P. Balandin [1]:

$$P = P_{ppr} \cdot e^{-K}$$

where P_{ppr} is prepressing pressure; K is the exponential factor:

$$K = -k_b \cdot f \cdot h / R,$$

here k_b is the coefficient of side pressure; f is friction coefficient; h is the depth of feedstock poured into the mold; R is the hydraulic radius of mold cross-section. In line with the equation, during pressing, the change in pressure and force depends only on variable h that in this case is determined by the piston position. Hence, the above conditions of increased pressing efficiency are directly connected with function S^* . They are connected with peculiarities of the changes in this function that were analyzed using an alternative of the pressing piston velocity (figure 2):

$$Vq = d(y^*_D(\varphi_1))/d\varphi_1 \quad (15)$$

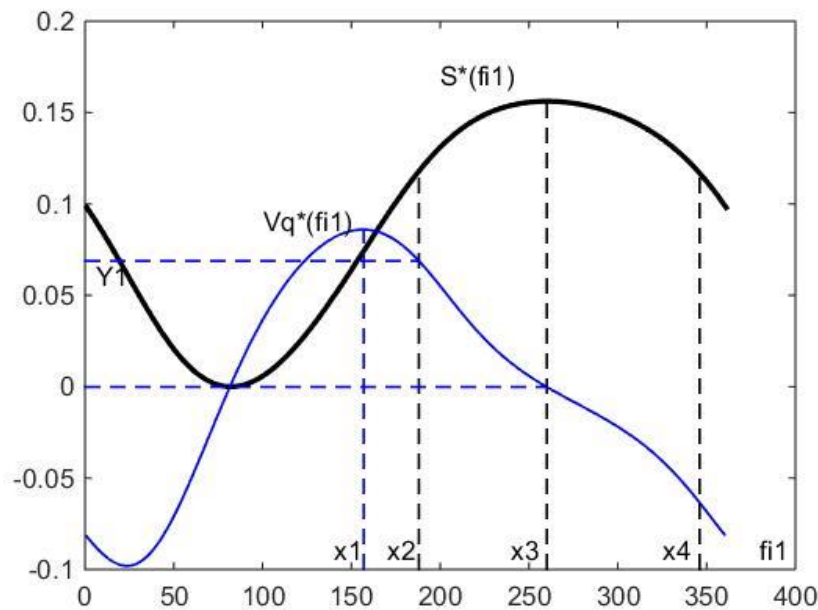


Figure 2. The kinematic functions of the pressing lever mechanism: $fi1$ is crank turning angle φ_1 ; $S^*(fi1)$ is function of position $S^*=y^*_D(\varphi_1)$; $Vq^*(fi1)$ is the alternative of the pressing piston velocity $Vq^*(\varphi_1)$; $Y1$ is $Vq^*(\varphi^*_{1beg})$; $x1$ is $\varphi_1=\varphi^*_{1Vmax}$; $x2$ is $\varphi_1=\varphi^*_{1beg}$; $x3$ is $\varphi_1=\varphi^*_{1m}$; $x4$ is $\varphi_1=\varphi^*_{1end}$.

The task of the analysis is to compare the calculation results of equation (15) for synthesized mechanism Vq^* and its prototype Vq . Due to complexities with implementation of integral estimates for the whole cycle, let us compare functions Vq^* and Vq only at two conditional stages connected with the beginning and the end of pressing.

At the first stage at $\varphi_1 \geq \varphi^*_{1beg}$, the more the piston velocity, the more the pressing process effectiveness. Thus, the best mechanism synthesis variant will be when

$$\varphi^*_{1Vmax} > \varphi^*_{1beg} \quad (16)$$

where φ^*_{1Vmax} is the crank turning angle at maximum Vq^* . If this condition is false, then let us use the alternative of velocity in the beginning of pressing as a comparison:

$$\delta_v = Vq^*(\varphi^*_{1beg}) - Vq(\varphi_{1beg}) \quad (17)$$

and crank turning angle in the course of which the velocity alternative will decrease down to the limit value V_{Rmax} :

$$\delta_{\varphi max} = |\varphi^*_{1beg} - \varphi^*_{1Rmax}| \quad (18)$$

where φ^*_{1Rmax} is the crank turning angle determined from equation:

$$Vq^*(\varphi^*_{1Rmax}) = V_{Rmax}$$

here V_{Rmax} is the limit value in fractions R_{max} of the velocity alternative in the beginning of pressing:

$$V_{Rmax} = Vq^*(\varphi^*_{1beg}) - Vq^*(\varphi^*_{1beg})/R_{max} \quad (19)$$

Parameter R_{\max} should be first selected for function $Vq(\varphi_1)$ of the prototype. While $\varphi_{1R\max}^*$ should not exceed 20% of $|\varphi_{1\text{beg}} - \varphi_{1m}|$, since high velocity is required only at the initial pressing stage.

The same estimation method will be used for the end stage of pressing. Then:

$$\delta_{\varphi\min} = |\varphi_{1\text{end}}^* - \varphi_{1R\min}^*| \quad (20)$$

where $\varphi_{1R\min}^*$ is determined from the following equation: $Vq^*(\varphi_{1R\min}^*) = V_{R\min}$, while the limit value in fractions R_{\min} of the velocity alternative from

$$V_{R\min} = Vq^*(\varphi_{1\text{beg}}^*) - Vq^*(\varphi_{1\text{beg}}^*)/R_{\min} \quad (21)$$

In this case, the rational solutions are determined by the combination of the maximum values of $\delta_{\varphi\min}$ and R_{\min} .

Besides, the possible synthesis of mechanisms with increased pressing duration should be taken into account. Then, the pressing phase takes the major part of range $\varphi_{1\text{beg}}^* \dots \varphi_{1\text{end}}^*$, and in the analysis of synthesis results is determined by the following condition:

$$\varphi_{1m}^* > \varphi_{1m} \quad (22)$$

Thus, to identify the region of rational results of the mechanism synthesis, one should program their comparison using criteria (16)...(22). This will allow selecting several best variants out of hundreds. The final choice can be made after their analysis at any attainable level of pressing machine operation detalization.

4. Conclusions

The results of the studies of the process of pressing silicate bricks impose new requirements on the pressing mechanism. To ensure them, in many cases, it is sufficient only to modernize the mechanism. Due to the complexity of this task, there are no generally accepted methods of its formalization and solution. However, it is possible to identify the individual operations that each researcher will have to perform. These include the models, conditions, and algorithms discussed in this paper. They demonstrate the possibility of building a computational process that will allow not only to find workable solutions, but also to select the most effective ones.

In the theory of mechanisms, this approach refers to the problem of multiparameter optimization in the framework of kinematic synthesis of a mechanical system. The peculiarity of the considered issue is that the required law of pressure changes during the pressing and the function of the press position do not allow one to form the target function. Therefore, the newer scientific information about the pressing process, the more justified the system of criteria for evaluating its effectiveness through modernization. This determines the direction of further researches in the field of improvement of the mechanical equipment for the production of silicate bricks.

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