2024. Volume 8, Number 1 **Pages 58-62**

UOT: 531/534.01:51-72
DOİ: https://doi.org/10.30546/09085.2024.106.128

MINIMIZING ENERGY CONSUMPTION WHEN TURNING ON AND STOPPING THE MAIN EXECUTIVE MECHANISM OF THE CPM

A.H. GULIYEV[1], B.H.ALIYEV[2], Y.A. ABDULAZIMOVA[3]

[1] *Baku Engineering University, "Mechanical Engineering" department, Baku, Azerbaijan*
asquliyev@beu.edu.az
[2] *Baku Engineering University, "Mechanical Engineering" department, Baku, Azerbaijan,*
baeliyev@ beu.edu.az
[3] *Baku Engineering University, "Mechanical Engineering" department, Baku, Azerbaijan,*
yabdulazimova@beu.edu.az

ARTICLE INFO

Article history:
Received: 2024-11-29
Received in revised form: 2024-12-02
Accepted: 2024-12-03
Available online

Keywords: Optimization, forging and pressing machine, optimality criteria, main executive mechanism, planetary drive.

ABSTRACT

The issues of minimizing energy consumption for turning on and off the main executive mechanism are considered. Analytical dependencies are obtained for calculating the optimality criteria depending on the drive parameters. This study explores methods to minimize energy consumption during the start-up and stopping phases of the main executive mechanism of a Continuous Processing Machine (CPM). These phases often involve significant power surges, contributing to energy inefficiencies and operational costs. The research investigates energy-efficient control algorithms, optimized motor drive systems, and dynamic scheduling techniques to enhance performance while reducing energy usage. A combination of theoretical modeling, simulations, and experimental validation demonstrates the effectiveness of the proposed approaches. The findings provide actionable insights for improving the energy efficiency of CPM systems, supporting sustainable manufacturing practices, and reducing operational expenses without compromising productivity or system reliability.

Introduction. The efficiency of the CPM (forging and pressing machine) as a whole and its individual units is possible with the optimal choice of its parameters, ensuring the highest technical and economic indicators of its operation.

One of the criteria for the optimality of the CPM is the minimization of energy consumption for turning on and off the GIM. The energy consumption for switching on and stopping the CPM with a planetary drive is determined by the formula:

$$A = I_a \omega_{an}^2 (j_n + j_0), \qquad (1)$$

where Ja - is the moment of inertia of the driving parts of the drive, reduced to the central (sun) gear of the gearbox;

ωaн - is the nominal angular velocity of the central gear;

jп, jO - are, respectively, the relative moments of inertia of the driven parts of the drive when switching on and stopping

$$J_n = \frac{J_b}{J_a p^2}, J_O = \frac{J_h}{J_a(1+p^2)} \qquad (1.1)$$

Jb - is the moment of inertia of the outer gear of the gearbox;

Jh - is the moment of inertia of the carrier, intermediate gear and CPM parts.

Substituting the values of the relative moments of inertia jп and jO, after simple transformations, the formula can be represented as,

$$A = \varpi_N^2 i_O^2 \left[\frac{J_v}{p^2} + \frac{J_h}{(1+p^2)}\right] \qquad (2)$$

where ωн - is the nominal angular velocity of the main shaft of the CPM.

The moments of inertia Jb and Jh- generally depend on the drive circuit (planetary gear type, presence of intermediate gear transmission), the transmitted load, the design of the drive components and the breakdown of the overall gear ratio iO, i.e. the parameters p and iZ.

Research methodology. It is practically impossible to obtain precise functional dependencies for determining the moments of inertia Jb and Jh due to the diversity and design complexity of the drive components. For comparative calculations, an approximate analysis method is used, according to which the moments of inertia Jb and Jh are defined as the moments of inertia of several main drive components (drive elements), represented as simple geometric figures. The design coefficients take into account the correspondence between the actual and calculated moments of inertia and the influence of other drive components, the moments of inertia of which are approximately taken as multiples of the moments of inertia of the drive elements. The moment of inertia Jb of a planetary gear drive is defined as the moment of inertia of a solid ring replacing a gear wheel with internal engagement. The inner diameter db and the width bb of the ring are taken to be equal to the initial diameter and the width of the gear ring, and the outer diameter of the wheel Db is a multiple of the diameter db, i.e. Db=dbβ, where β is the proportionality coefficient.

Expressing the width bb of the gear ring through the diameter db and the relative width of the ring ψb, the moment of inertia Jb is determined by the formula

$$J_b = \frac{\pi l}{32} k_b b_b \left(D_b^4 - d_b^4\right) \frac{\pi l}{32} k_b \psi_b k_b d_b^5 \qquad (3)$$

were

The diameter db is determined by the condition of the contact strength of the working surfaces of the teeth

$$d_b = d_b = p \sqrt[3]{\frac{2M_{sm}}{pi_Z(p-1)\psi_b n_\varpi [K_0]_R}} \qquad (4)$$

Substituting the values db into formula (3) and denoting

$$E_b = \frac{\pi\gamma}{32} k_b \psi_b k_b \beta \sqrt[3]{\frac{2}{\psi_b n_\varpi}}$$

we obtain a dependence for determining the moment of inertia Jb in the form

$$J_b = E_{b\sqrt[3]{\left(\frac{M_{sm}}{[K_0]_R}\right)^5}} \left(p \sqrt[3]{\frac{1}{pi_Z(p-1)}}\right) \qquad (5)$$

The moment of inertia Jh is the sum of the moments of inertia of the carrier and the reduced moment of inertia of the intermediate gear

$$J_h = J_{hR} + J_{hZ}$$

If the carrier is represented as a solid cylinder with dimensions Dh and dh multiples of the dimensions of the gear engagement of the wheel b, i.e., Dh = kddb, bh = kbbb the moment of inertia JhR can be determined by the formula

$$J_{hR} = \frac{\pi\gamma}{32} k_h b_h D_h = \frac{\pi\gamma}{32} k_h k_\beta k_d^4 \psi_d d_b^5 \ (6)$$

where kd, kb are proportionality coefficients.

Substituting the value of db according to formula (4) and designating

$$E_{hR} = \frac{\pi\gamma}{32} k_h k_d^4 \psi_b \left(\sqrt[3]{\frac{2}{\psi_b n_\varpi}}\right)^5$$

formula (6) we represent as

$$J_{hR} = E_{\{hR} \sqrt[3]{\left(\frac{M_{sm}}{[K_0]_R}\right)^5} \left(p \sqrt[3]{\frac{1}{p i_Z (p-1)}}\right)^5 \quad (7)$$

The moment of inertia JhZ is defined as the moment of inertia of the gear wheels of the intermediate gear reduced to the driven shaft of the planetary gear reducer, represented in the form of solid disks with dimensions equal to the dimensions of the toothed rims of the corresponding wheels and parts of the GIM. For a single-stage gear transmission with the same width of the toothed rims of the pinion and wheel, the moment of inertia JhZ is determined by the formula

$$J_{hZ} = \frac{\pi\gamma K_G}{32} b_1 (k_1 d_1^4 + k_2 d_2^4 i_Z^{-2}) = \frac{\pi\gamma K_G}{32} \psi_2 i_Z d_1^5 (k_1 + k_2 i_Z^2) \quad (8)$$

The coefficient KG takes into account the magnitude of the moment of inertia of the GIM parts in the total moment of the driven masses. Its average value is 1.2-1.4.

The diameter of the gear d1 is usually limited by the contact strength of the working surfaces of the teeth, therefore, using the corresponding dependencies, formula (8) can be represented as

$$J_{hZ} = E_{hZ} b \sqrt[3]{\left(\left(\frac{M_{sm}}{[K_0 1]_R}\right)^5\right)} \left(\sqrt[3]{i_Z + 1}\right)^5 (k_1 + k_2 i_Z^2) i_Z^{-4} \quad (9)$$

Were

$$E_{hZ} = \frac{\pi\gamma}{32} \psi^2 \left(\sqrt[3]{\frac{2}{\psi^2}}\right)^5$$

If the gear engagements of the planetary gearbox and the intermediate gear are made of equal strength (i.e., assume that $[K_0]_R = [K_0]_Z$), , which ensures both the lowest total weight of the drive and its lowest inertia, the energy consumption for switching on and stopping can be determined by the formula

$$A = \varpi_N^2 i_O^2 (1+p)^n \varpi_H^2 \left(\frac{M_{sm}}{[K_0]_R}\right)^5 \left[+\frac{E_{hZ}}{(1+p)^2}\left(\sqrt[3]{i_Z+1}\right)^5 (k_1 + k_2 i_Z^2) i_Z^2 i_Z^2\right) \quad (10)$$

The value $\varpi_H^2 \left(\sqrt[3]{\frac{M_{sm}}{[K_0]_R}}\right)^5$ does not depend on the drive parameters and is constant for each CPM. Therefore, the ratio

$$R_A = \frac{A}{\varpi_H^2}\left(\sqrt[3]{\frac{[K_0]_R}{M_{sm}}}\right)^5 \quad (11)$$

(11) represents the dependence of the relative energy consumption for switching on and stopping the GIM on the drive parameters and is taken as the second criterion for the optimality of the RA parameters.

Discussion of results. The design coefficients of the gear wheels and the proportionality coefficients are taken equal to:

kb = 1. 3..2.4; kh = 1. 08..1.85; k1 = 1. 4..2.5; k2=0. 27..0.42; β=1. 1..1.17; kd=0. 7..0.82. For relatively narrow gear wheels (ψb≤0.15), smaller values of the coefficients kd, β and larger values of the coefficients kb, k1 k2 are adopted.

Fig. 1, a) shows the dependence of the RA criterion on the parameters p and iZ, and Fig. 1, b) on the parameter p for a certain general gear ratio p and iO. The graphs in Fig. 1, a) show the degree of influence of each of the parameters p and iZ on the value of energy consumption A. From Fig. 1, b it is clear that for each gear ratio iO there is a unique combination p and iZ at which the value of RA will be the smallest. The optimal values of the parameter p are found as the coordinate of the intersection point of the RA curve for the corresponding value of iO with the line SA, which is the line of minimum values of the RA criterion.

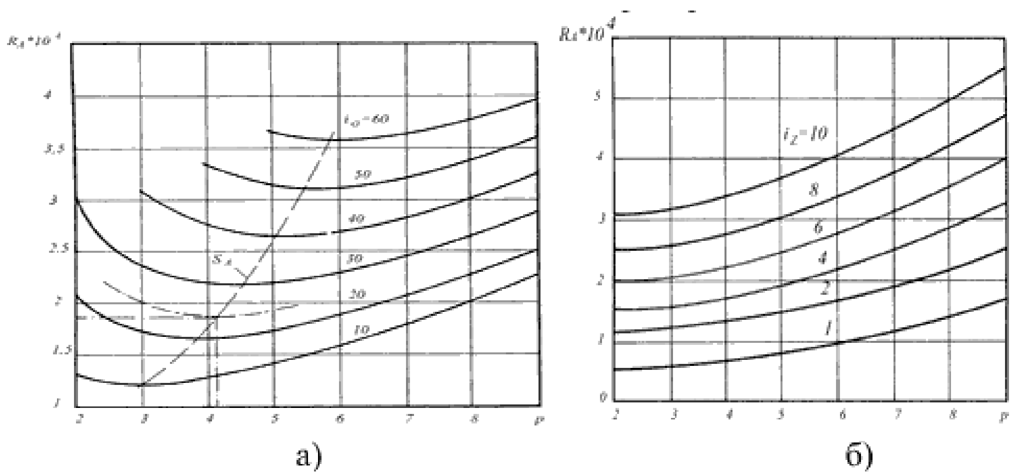

Fig. 1. Dependence of the RA criterion on the drive gear ratios

According to Fig. 1, b), it is also possible to estimate the degree of overestimation of energy consumption compared to the minimum value with non-optimal drive parameters. For example, with

iO = 25, the lowest energy consumption will be at p = 4.12 and iZ = 4.88 (shown in Fig. 2 by a dash-dotted line). If we take p = 3 and iZ = 6.25, the energy consumption will be overestimated by 1.06 times.

Conclusion

1. Thus, analytical dependencies for calculating the optimality criteria depending on the drive parameters were obtained.
2. This study has demonstrated effective strategies to minimize energy consumption during the start-up and stopping phases of the main executive mechanism of a Continuous Processing Machine (CPM). By implementing optimized control algorithms, energy-efficient drive systems, and operational protocols, significant energy savings can be achieved without compromising performance.
3. The experimental and simulation results confirm the potential of these approaches to enhance efficiency and sustainability. These findings contribute to reducing the environmental footprint of CPM systems and provide practical insights for industries seeking to optimize energy use, lower operational costs, and promote environmentally responsible manufacturing practices. Future work could explore real-time adaptive systems for further improvements.

REFERENCES

1. Yavtushenko A.V., Rudenko A.V., Rybinok V.A. Improving the systems for engaging crank forging and pressing machines. – K.: Knowledge, 1980. – 24 p.
2. Tynyanov V. N. On designing gear drives with the least mass for crank presses // Forging and stamping production – 1976, No. 9, – P. 31-34. 3. Planetary gears. Handbook. Ed. V. N. Kudryavtsev and Yu. N. Kirdyashev. – L.: Mechanical engineering, 1977. – 536 p.