IMPROVING THE PROCESS OF PILGER ROLLING BASED ON THE CHOICE OF RATIONAL PARAMETERS OF FORHOLLER

CHECHULIN Yury B.[1], POPOV Yury A.[1], SOKOLINSKY Boris Y.[2], GAAS Evgeny A.[2]

[1]*Ural Federal University, Yekaterinburg, Russian Federation, decanat@mmf.urfu.ru*
[2]*JSC «CHELYABINSK PIPE-ROLLING PLANT», Chelyabinsk, Russian Federation, pavel.sokolinskiy@chelpipe.ru*

Abstract

For the newly proposed pneumohydraulic drive of workpiece return movement it is researched relations, analysis was carried out and recommendations of selecting values of the initial pressure in the pneumatic chamber, ensuring a rational ratio between the periods of acceleration and deceleration were obtained. The mathematical model and program were created for calculation of the law of motion of the brake rod with floating piston device using the system variable throttling the flow of the working fluid along the movement in the brake chamber. On the basis of the developed model there is made the numerical analysis to assess the impact of the inner surface of the brake axle box on the final rate of the brake piston, as well as the effect of the location and change in the area of additional throttling apertures therein on the trend values of the maximum braking acceleration of moving masses. There is obtained the algorithm of determining the constants and variables control law of motion, minimizing the maximum values of dynamic loads per cycle movement of the workpiece. It is shown that the proposed embodiment of the rod system of the workpiece return movement in the working cage with a floating piston ensures reduce the moving masses and increase the rigidity of the movable rod. It is obtained calculated relations for the definition of effort mobile carriage prop, providing gapless its support with mechanism of feed limit.

Keywords: Pneumatic and brake chamber, floating piston, variable throttling, mechanism of feed limit

MAIN TEXT

Tube units with Pilger rolling mill are used for the manufacture of thick-walled seamless pipes of large diameter of carbon and alloy steels. The rolls of fixed working stand provided with streams of variable radius periodically deform the next portion of the workpiece on the movable metal mandrel. When the working rolls during each revolution of the workpiece together with the deformable mandrel moves cyclically in the last length (1500 mm) in the direction opposite to the exit of the finished tube mill. In the final part of the stream is expanded to a radius greater than the radius of the workpiece, for its return to the free zone and the additional deformation in the direction of displacement of the output value of the finished pipe to a single feed. At the same time it made the workpiece rotate around its axis at an angle tilting (up to 90°).

Forholler operates in severe dynamic loading mode. Joint mass of the workpiece and the mandrel reaches 10 tons. The duration of the return tube blank in compression zone, including the acceleration and deceleration is less than 1/3 of a full cycle of a single rolling. At the same time the need for a return after acceleration reliable braking of moving parts to a predetermined position of the beginning of the working portion of the stream in a short period of time (duration from 0.15 to 0.3 seconds for the main mill sizes) there are big acceleration - the source of the high inertial forces. Therefore, the main objective of increasing efficiency Pilger rolling mill is to further improve the design and supply the machine settings [1].

All units of the existing mills liner supply rolls is carried out in a pneumatic drive and braking can be done in many ways - with pneumatic, hydraulic or mechanical devices. Recent forhollers with hydraulic brake apparatus are widespread received.

As the object of modernization there was selected forholler of pilger rolling mill №2 of Chelyabinsk Pipe Rolling Plant, which uses mechanical braking blank (**Figure 1**). Its construction includes a housing with an integrated pneumatic cylinder in it, brake chambers, tilting drill pipe, articulated with a piston and a ratchet device. Drill guide bushings is mounted in rotatable about the longitudinal axis. Brake chamber is a pack of Belleville springs, on the one hand limited damper, and on the other hand - flange cap.

The current design has a number of drawbacks. Since the work of the forholler is the nature and impact of its components are subjected to high dynamic loads, spring brake chamber quickly fail. In addition, it is not always possible to produce the spring to the desired heat treatment regime which reduces their stiffness and accelerates the destruction. After the destruction of the springs dynamic load is transmitted to the body of the air piston and the flange on the cover, causing periodic breakdowns. Another disadvantage of this design include the lack of control of the braking process.

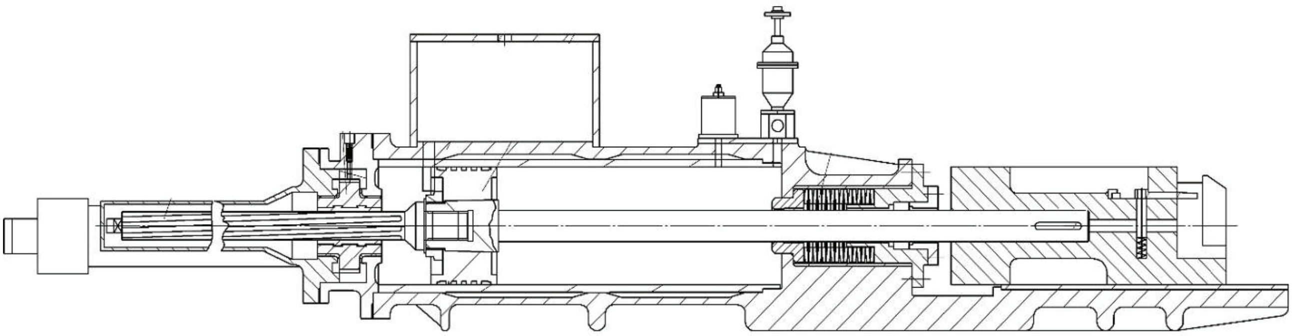

Figure 1 Construction of forholler with mechanical braking

In order to increase efficiency offered streamline design without changing the dimensions of the replacement of the spring brake chamber on a floating hydraulic piston. **Figure 2** shows the newly proposed structure.

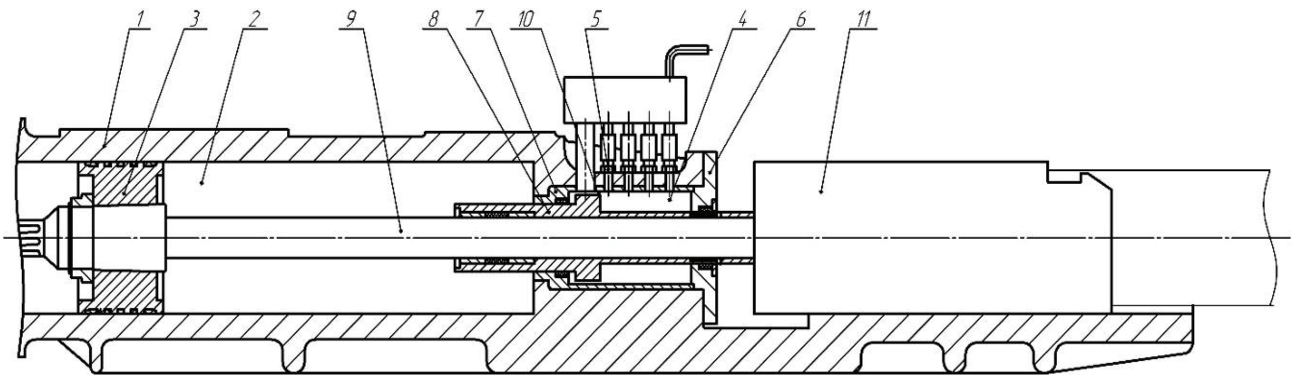

Figure 2 Construction of forholler with hydraulic braking:
1 - forgoller housing; 2 - air chamber; 3 - air piston; 4 - hydraulic chamber; 5 - throttling openings; 6 - flange cover; 7 - brake hub; 8 - hydraulic piston; 9 - stock; 10 - annular slot; 11 - arbor lock.

The advanced design allows you to control the braking mode depending on the size of the rental. As the working fluid chamber is provided braking running water having a lower heat capacity as compared with oil and improves the conditions of removal of excess heat. The brake chamber housing mounted brake hub, which on one side rests against the back support assembly, and on the other - is supported flanged lid. The hydraulic piston is designed as a bushing on stem planted to be freely movable therealong. When the return movement of the piston rod of the air contact with the piston sleeve hydraulic piston divides the workpiece between the rolls in the acceleration and deceleration of the moving masses, including arbor lock mandrel (mandrel) and the rolled product. Braking is performed by the hydraulic resistance of the expiration of the fluid through the annular gap and throttling holes. A distinctive feature of the devices of this type is a two-fold reduction in the mass of the drive arbor lock.

An urgent task for this type of brake device is to determine the constant parameters of the brake chamber defined by design variables and control parameters, which make it possible to adjust the braking mode for a certain size of the rolled tubes.

Acceleration period of the lower end position is carried out with compressed air in the pneumatic chamber [2]. For this stage of the calculation of the equation of motion can be written as

$$m \cdot \ddot{x} - p_0 \cdot F_1 \left(\frac{H}{H-h+x}\right)^k + R = 0, \tag{1}$$

where

m - the mass of the moving parts of the machine and the sleeve;

F_1 - active area of the air piston;

p_0 - the initial air pressure in the pneumatic chamber;

H - length of the air space in the most forward position of the piston, the piston reduced;

h - rollback of the piston during the rolling (piston stroke);

R - friction force,

k - adiabatic index for air.

The result of the solution of equation (1) is the value of time and the final acceleration rate, which is the initial condition for the braking phase.

In the period of active hydraulic brake piston, passing along the brake sleeve with known dimensions of the brake chamber equation of motion becomes [3], [4]:

$$m \cdot \ddot{x} + p_0 \cdot F_1 \left(\frac{H}{H-l+x}\right)^k - p_H \cdot F_2 - R = 0, \tag{2}$$

where p_H - changing the pressure in the hydraulic chamber during braking [5].

$$p_H = \frac{12 \cdot \mu \cdot Q \cdot l'}{b \cdot (\delta + \frac{f}{b})^3} + \frac{\rho \cdot Q^2}{2 \cdot b^2 \cdot (\delta + \frac{f}{b})^2} \tag{3}$$

Where

$Q = F_2 \cdot \dot{x}$ - the total flow rate through the annular slot and throttling orifice;

μ - dynamic viscosity of the fluid;

l' - the length of the annular gap;

F_2 - the current value of the area of the annular gap,

δ - the size of the annular gap,

f - total area of the throttling openings,

b - the perimeter of the annular gap [6].

The first term of equation (2) represents the pressure loss due to friction in the fluid, and the second hydrodynamic losses. From the joint decision (3) (2), we obtain:

$$m \cdot \ddot{x} + p_0 \cdot F_1 \left(\frac{H}{H-l+x}\right)^k - \frac{12 \cdot \mu \cdot F_2^2 \cdot \dot{x} \cdot l'}{b \cdot \left(\delta + \frac{f}{b}\right)^3} - \frac{\rho \cdot F_2^3 \cdot \dot{x}^2}{2 \cdot b^2 \cdot (\delta + \frac{f}{b})^2} - R = 0, \tag{4}$$

Consider the acceleration and deceleration phases of the above forgoller the following parameters:

The main criteria for the rational decision to choose a minimum speed of movement of the rod at the end of its return movement, as well as the minimum of the maximum pressure in the brake chamber and maximum acceleration. We represent the solution of the equation (4) to the mobile mass of 10000 kg with various configurations of brake hub cone and at different throttling. Other parameters according to **Table 1** and **2**. The initial pressure of the air chamber of 1.5 atm. (The minimum value at which the sum of the acceleration and deceleration time does not exceed the permissible value of 0.5).

Table 1 Constant conditions

Characteristic	Symbol, unit	Numerical value
The active area of the air piston	F_1, m²	0.298
Piston stroke	h m	1.55
The area of the hydraulic piston	F_2, m²	0.101
The diameter of the hydraulic piston	D_H, m	0.437
The diameter of the supporting rod	D_S, m	0.250
The diameter of the hydro sleeve on the cylindrical portion	D_2, m	0.439
The length of the brake chamber	l, m	0.355

Table 2 Variable conditions

Characteristic	Symbol, unit	Numerical value
Weight of moving parts unit	m_{np}, kg	8000, 10000, 12000
The initial air pressure in the pneumatic chamber	p_0, $10^5 \cdot$Pa	1.2 - 2.0
Number of rows of holes in the brake chamber	n, pcs	max 4
The number of holes in the 1st row	n_1, pcs	max 2
The number of holes in the 2nd row	n_2, pcs	max 2
The number of holes in the 3rd row	n_3, pcs	max 2
The number of holes in the 4th row	n_4, pcs	max 2

Figure 3 shows the effect of the initial size of the annular gap to the hydraulic chamber of the braking. Changing this setting without affecting the value of the final speed, it affects the value and location of peak pressure and acceleration. When row arrangement of apertures throttling assigned on a "2-1-0-0" (in the first row of open holes 2, in the second - 1 hole, closed the third and fourth holes), hole diameter - 30 mm. For a given constant parameters (**Table 1**) in the most rational value of the initial size of the annular gap 4.5 mm.

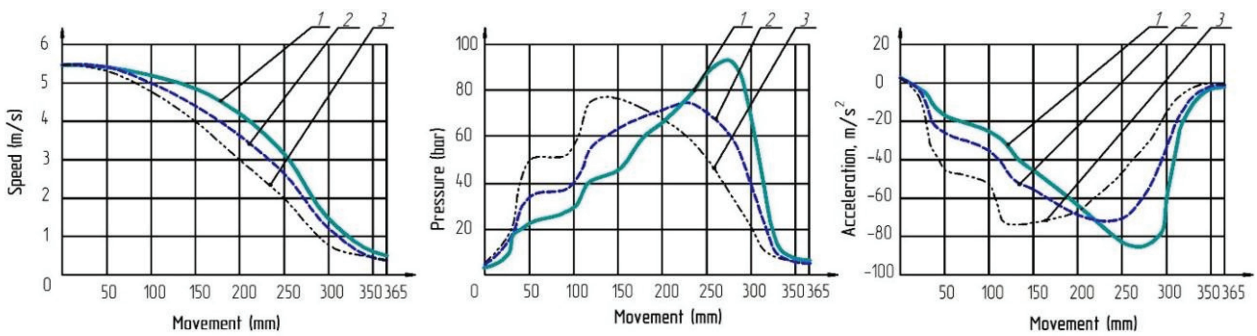

Figure 3 Drawing speed, pressure, and acceleration of movement.
1 - The curve with the initial size of the cone 5.5 mm; 2 - The curve with the initial size of the cone 4.5 mm; 3 - The curve with the initial size of the cone 3.5 mm.

Figure 4 shows the effect of the finite size of the annular gap in the braking process. This option allows to reduce the value of the final speed, and to a lesser extent, affects the maximum values of pressure and acceleration. As follows from the results of comparison of the final size reduction of the annular gap from 1.5 mm to 0.5 mm allows to reduce the final speed of the rod 6 times with increasing pressure and maximum deceleration by 20 %. **Figure 5** shows the effect of throttling on the braking characteristics.

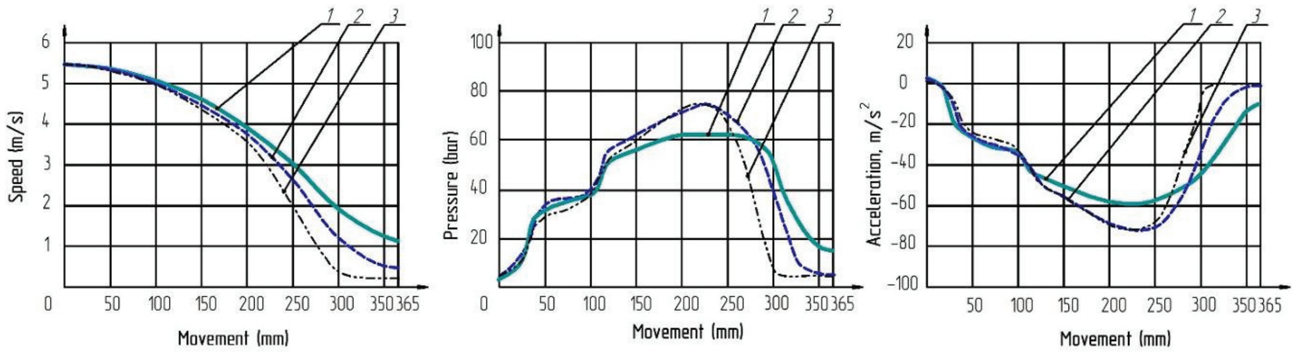

Figure 4 Drawing speed, pressure, and acceleration of movement:
1 - The curve at the end of the cone size of 1.5 mm; 2 - The curve at the end of the cone size of 1.0 mm; 3 - The curve at the end of the cone size of 0.5 mm.

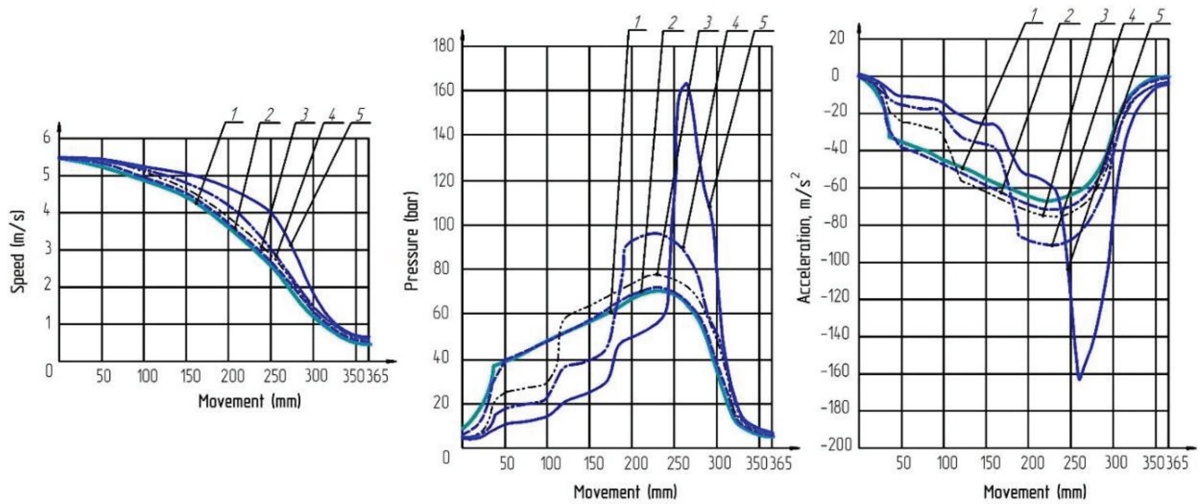

Figure 5 Drawing speed, pressure, and acceleration of movement:
1 - The curve throttling scheme 0-0-0-0; 2 - The curve throttling scheme 2-0-0-0; 3 - The curve throttling scheme 2-2-0-0; 4 - The curve throttling scheme 2-2-2-0; 5 - The curve throttling scheme 2-2-2-2.

The values of the initial and final dimensions of the annular gap adopted respectively 4.5 mm and 0.5 mm. In order to identify trends in the basic characteristics of the braking process, depending on the nature of the change in the course of the throttling piston brake adopted discrete version of the series arrangement of throttling apertures. As a result, it is established:

1) throttling does not affect the final rate of deceleration;
2) uniform throttling apertures opening throughout the length of the brake hub reduces the initial front brake chamber pressure, but its value does not exceed half the maximum value subsequently;
3) with the number of rows of openings along the length of the brake sleeve substantially increase the maximum pressure and accelerations;
4) most preferred for stable rolling process is a variant of an additional choke in the first third of the length of the stroke of the piston sleeve of the hydraulic chamber. Setting the acceleration and deceleration allows for optimal operation of the feeding device for the workpieces of different masses [7].

Setting the acceleration and deceleration parameters allows for optimal operation of the feeding apparatus for different masses blanks.

As a result of research aimed at reducing the dynamic loads on the mechanical system at the beginning of braking, the brake hub design offered with a monotonic change in the total area of the gap, due to the taper bushings and the location of throttling apertures along a helical path. **Figure 6** shows graphs speed, acceleration and pressure depending on the movement of the brake piston along the hydraulic chamber for this embodiment.

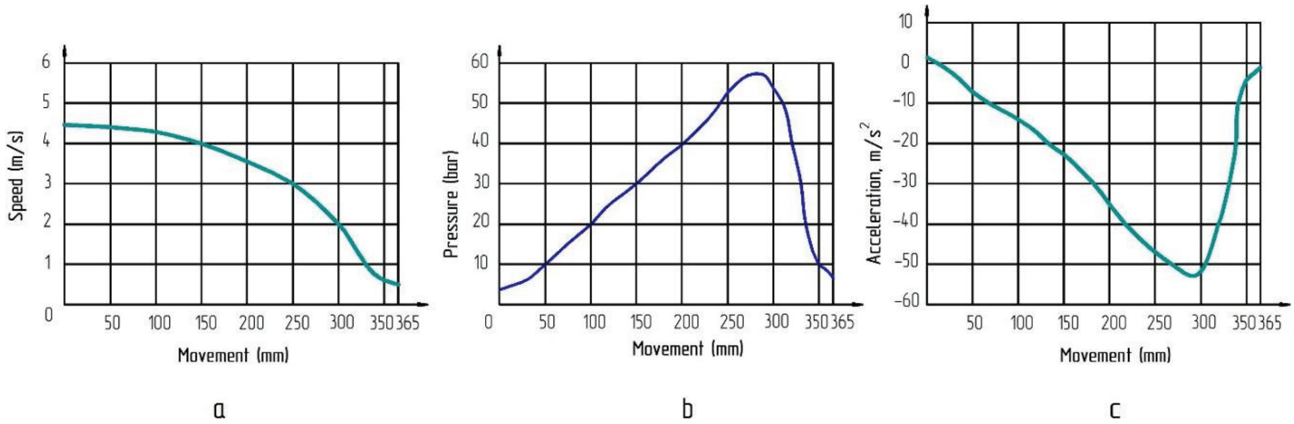

Figure 6 Drawings: a - Drawing speed; b - Drawing pressure; c - Drawing acceleration

This design allows the sleeve to receive a monotonic change in pressure and acceleration of movement, reducing fluctuations in forces affecting the grip preform mandrel. Charts show that the pressure build-up front and, respectively, and the acceleration does not contain discrete jumps. The maximum value of the pressures and the accelerations do not exceed 5 % of the values using hydraulic sleeve having a tapered profile.

CONCLUSION

The design of the forholler of the hot pilger rolling mill with shorter dimensions of the stem and the length of the brake chamber. This allows a greater longitudinal rod dimensions and the stability of the feeding apparatus and reduce the weight of moving parts dynamic loads.

An algorithm and a work program for calculating the parameters of the proposed mechanism, providing a choice of rational characteristics of service design and process control.

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