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POWER EQUIPMENT

Updating the Ten-Roller Screens in Coke Shop 1 at Ural Steel

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Abstract—In coke shop 1 at Ural Steel, double ten-roller screens are employed for primary sorting of coke to isolate the >25 mm fraction, which is sent by conveyers to large-coke hoppers for subsequent loading on railroad cars. These screens are of complex design and difficult to maintain and repair, while the drives are of low efficiency. In screen operation, unplanned downtime is encountered. To improve their reliability, the existing drives of the sorting stands and the mechanisms for moving the screen are replaced by simpler and more reliable designs with a gear motor. Updating the screens simplifies their design and improves their reliability; reduces maintenance and repair costs and the frequency of unplanned downtime; and lowers production costs.

Keywords: ten-roller screen, blast furnace, coke sorting, drives, gear motor, updating

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INTRODUCTION

In coke shop 1 at Ural Steel, double (two-stand) ten-roller screens are employed in the production of coke and its transportation to the blast furnace shop. Beyond coke ramps 1 and 2 (coke batteries 1, 3, and 4) and 3 (batteries 5 and 6), coke is sent by conveyers K-1a, K-2a, K-1b, K-2b (for first sorting) and K-16a and K-16b (for second sorting) to the ten-roller screens for primary isolation of the >25 mm fraction, which is sent by conveyers K-8 and K-19 to large-coke hoppers for subsequent loading on railroad cars [1].

The results of blast furnace operation depend on the quality of the coke, which is determined by the quality of the initial coal and its transportation [2, 3]. The economic performance of the blast furnaces and the whole enterprise depends directly on uninterrupted operation of the double ten-roller screens.

These screens are of complex design and difficult to maintain and repair, while the drives are of low efficiency [4–8]. In screen operation, unplanned downtime is encountered, on account of their poor reliability. In this context, reliability encompasses fault-free operation, durability, ease of repair, and retention of performance [9].

In the present work, the reliability of the ten-roller screens in coke shop 1 at Ural Steel is increased so as to minimize unplanned downtime and repair costs, with consequent decrease in the costs of coke production.

ANALYSIS

The double ten-roller screen consists of trolley *1*, on which are mounted two stands *8* for sorting the coke: the working stand and the backup (Fig. 1). The stand consists of ten parallel working rollers with projecting disks. The rollers are supported by roller bearings on a rigid frame, in a plane inclined at 15° to the horizontal. The design specifies that the speed of the rollers in the stand becomes higher on moving down the slope—that is, in the direction of coke motion. The rollers with projecting disks form a sorting surface with apertures whose shape and size are determined by the distance between the rollers and the shape of the disks.

Each stand has its own drive, consisting of electric motor *2*, elastic bush–pin coupling *3*, conical–cylindrical gear system *4*, gear-type coupling *5*, gearbox *6*, and equalizing coupling *7*. Trolley *1* consists of a welded frame mounted on six wheels, of which the center two are driven, while the others are not. The mechanism for moving the screen is located in the middle of the frame and consists of an electric motor, couplings of bush–pin and gear type, a cylindrical two-stage gear system, and open gears. The gears of the open transmissions are mounted on the hubs of the drive wheels.

The screen employs a centralized lubrication system for the gearboxes, consisting of plunger pumps, oil flow indicators, and pipelines. The oil is drawn in from the gearbox housings and sent in pipelines through the lids to the upper gears and bearings of the

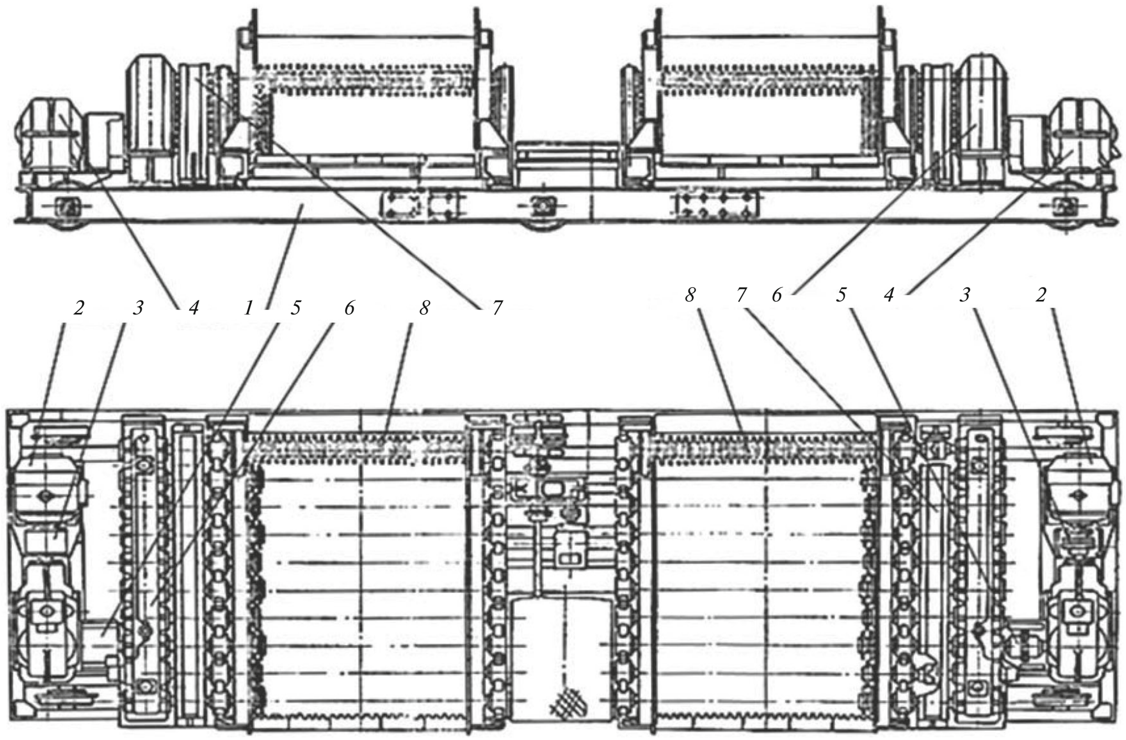

Fig. 1. Double ten-roller screen: (*1*) trolley; (*2*) electric motor; (*3*) elastic bush–pin coupling; (*4*) conical–cylindrical gear system; (*5*) gear-type coupling; (*6*) gearbox; (*7*) equalizing coupling; (*8*) stands.

gearboxes. Then it flows down and lubricates the other gears and bearings. Table 1 presents the characteristics of the double ten-roller screen.

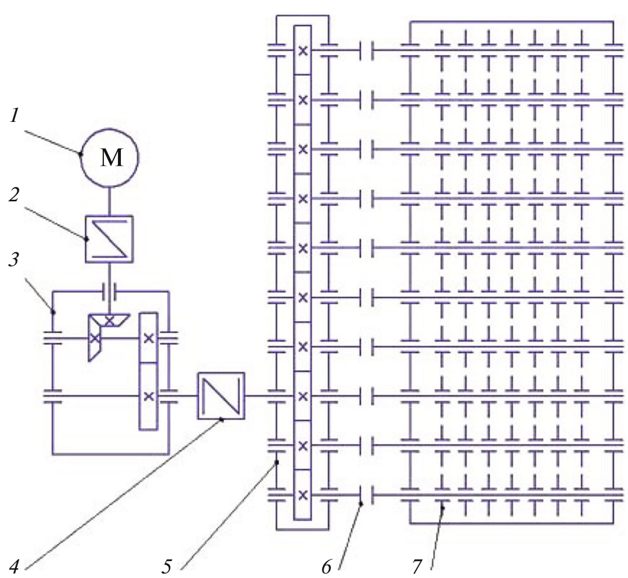

Fig. 2. Kinematic diagram of the drive for the coke sorting stands: (*1*) electric motor; (*2*) protective clutch; (*3*) conical–cylindrical gear system; (*4*) gear-type coupling; (*5*) gearbox; (*6*) equalizing coupling; (*7*) stands.

The coke sent to the upper part of the working stand is agitated by the rotating disks and rolls at increasing speed down the slope formed by the disks. The coke fines fall into the square cells formed by the disks, and the larger coke pieces fall off the screen. A kinematic diagram of the drive for the sorting stands is shown in Fig. 2.

The speed of the gearbox's low-speed shaft (output shaft) is

$$n_{\mathrm{ls}} = n_{\mathrm{mo}}/u_{\mathrm{g}} = 970/20 = 48.5 \text{ rpm},$$

where $n_{\mathrm{mo}} = 970$ rpm is the shaft speed of the AO72-8 electric motor in the drive of the sorting stand; and $u_{\mathrm{g}} = 20$ is the gear ratio of the KTs1-300 conical–cylindrical gear system in the stand drive.

The power at the low-speed (output) shaft of the drive for the sorting stands is

$$P_{\mathrm{ls}} = P_{\mathrm{mo}}\eta_{\mathrm{co}}^2\eta_{\mathrm{g}} = 10 \times 0.982 \times 0.93 = 8.93 \text{ kW},$$

where $P_{\mathrm{mo}} = 10$ kW is the power of the AO72-8 electric motor in the drive of the sorting stands; $\eta_{\mathrm{co}} = 0.98$ is the efficiency of the coupling [11, 12]; and $\eta_{\mathrm{g}} = 0.97 \times 0.96 = 0.93$ is the efficiency of the KTs1-300 conical–cylindrical gear system in the stand drive, with the bearings [11, 12].

The torque at the shaft of the AO72-8 electric motor is

$$T_{mo} = (P_{mo}/\omega_{mo}) \times 10^3$$
$$= 10 \times (10^3/101.5) = 98.5 \text{ N m},$$

where $\omega_{mo} = \pi n_{mo}/30 = (3.14 \times 970)/30 = 101.5$ rad/s is the speed of the output shaft of the AO72-8 electric motor in the drive of the sorting stand.

The torque at the low-speed shaft (output shaft) of the gear system is

$$T_{ls} = T_{mo} u_g \eta_{co}^2 \eta_g = 98.5 \times 20$$
$$\times 0.98^2 \times 0.93 = 1759.5 \text{ N m}.$$

To increase the reliability of the screens, reduce their unplanned downtime, and minimize the time lost to repairs, the existing drive of the coke sorting stands, which contains an electric motor, a gear system, and two couplings, must be replaced by a simpler and more reliable design including a gear motor. That significantly simplifies maintenance and repair of the screen, with corresponding decrease in the time and effort required [10]. A kinematic diagram of the updated drive for the coke sorting stands is shown in Fig. 3.

For the updated drive, we employ a K77DV160M4 conical–cylindrical gear motor with the following characteristics: motor power $P_{mo} = 11$ kW; shaft speed of motor $n_{mo} = 1431$ rpm; and speed of gear motor's output shaft $n_{sl} = 62$ rpm.

The gear ratio of the drive is

$$u_{gm} = n_{mo}/n_{ls} = 1431/62 = 23.08.$$

The torque at the motor shaft is

$$T_{mo} = P_{mo}/\omega_{mo} = 11000/149.8 = 73.4 \text{ N m},$$

where $\omega_{mo} = \pi n_{mo}/30 = (3.14 \times 1431)/30 = 149.8$ rad/s is the speed of the electric motor's output shaft.

The torque at the output shaft of the gear motor is

$$T_{ls} = T_{mo} u_{gm} \eta_{gm} \eta_{co}$$
$$= 73.4 \times 23.08 \times 0.94 \times 0.98 = 1560.5 \text{ N m}.$$

The calculated torque for the coupling is

$$T_{ca} = K_1 K_2 T_{sl} = 1.2 \times 1.5 \times 1560.5 = 2808.9 \text{ N m},$$

where the constant $K_1 = 1.2$ takes account of the contribution of the transmission; and the constant $K_2 = 1.5$ takes account of the coupling's working conditions.

We select an MZ-3-∅50/∅55 coupling with maximum torque $T_r = 3150$ N m at the maximum speed $n_{max} = 4000$ rpm.

A check calculation to prevent impermissible wear rates gives the result

$$p = \frac{T_{ca}}{0.9bd^2} = \frac{2808.9 \times 1000}{0.9 \times 20 \times 120^2} = 10.8 \text{ MPa},$$

where $b = 20$ mm is the tooth length; $d = m \times z = 3 \times 40 = 120$ mm is the diameter of the bush's splitting circumference; $m = 3$ is the engagement module, mm; and $z = 40$ is the number of teeth for the bush.

The permissible crumpling stress is $[p] = 12$–15 MPa. The result $p = 10.8$ MPa is less than this value and so the characteristics of the coupling are satisfactory.

A significant defect of the double ten-roller screens is that the drive of the mechanism moving the screens breaks down fairly often, on account of wear of the open gears and the closed gears in the cylindrical gear system. Such wear results from the complex design of the drive and its challenging operating conditions, in

Table 1. Characteristics of ten-roller screens

Characteristic	Value
Productivity (throughput of a single stand), t/h	175
Roller speed, rpm	63–86
Speed of rotating disks, m/s	0.5–0.9
Speed of screen motion, m/s	0.06
Roller inclination, deg	15
Number of rollers	10
Width of stand, mm	1650
Length of stand, mm	2214
AO72-8 electric motor of stand drive:	
power, kW	10
speed, rpm	970
KTs1-300 gear system of stand drive:	
gear ratio	20
torque at low-speed shaft	2100
MTKN 12-6 electric motor of mechanism for moving screen:	
power, kW	3
speed, rpm	905
RTsD-250 gear system of mechanism for moving screen:	
gear ratio	40
rated torque at low-speed shaft, N m	1292
Open gear:	
module	6
number of gear teeth	72
number of teeth on pinion shaft	11
gear ratio	6.54
Mass, kg:	
without electrical equipment	26874.0
with electrical equipment	27378.3
Screen dimensions, mm:	
length	8200
width	2750

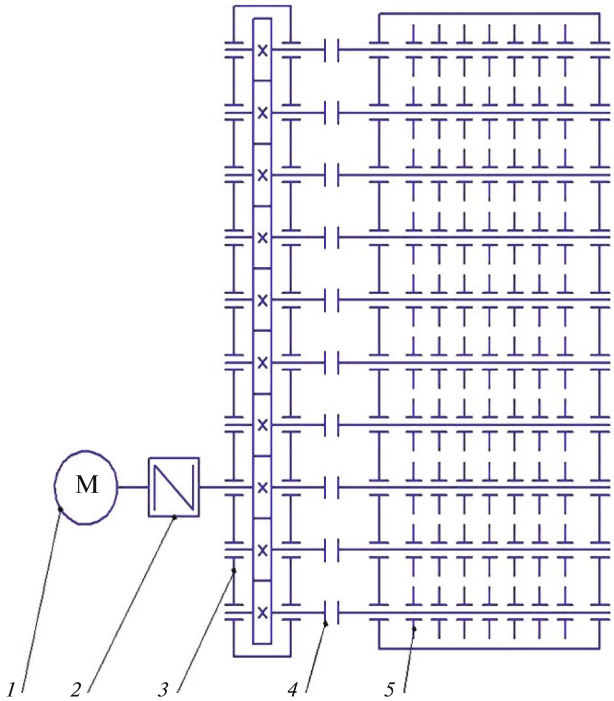

Fig. 3. Kinematic diagram of the updated drive for the coke sorting stands: (*1*) gear motor; (*2*) gear-type coupling; (*3*) gearbox; (*4*) equalizing coupling; (*5*) stands.

that the open gears operate in an atmosphere with a high dust content. The overall efficiency of the drive responsible for moving the screens is low (Fig. 4), and its repair and maintenance costs are increased.

The overall efficiency of the drive responsible for moving the screens is

$$\eta = \eta_{co}^3\eta_{cl}^2\eta_{op}^2\eta_p^4 = 0.98^3 \times 0.97^2 \times 0.96^2 \times 0.99^4 = 0.78,$$

where $\eta_{co} = 0.98$ is the efficiency of the coupling [11, 12]; $\eta_{cl} = 0.97$ is the efficiency of a single closed cylindrical gear system with the bearings [11, 12]; $\eta_{op} = 0.96$ is the efficiency of a single open cylindrical gear system with roller bearings, in the presence of plastic lubricant [11, 12]; and $\eta_p = 0.99$ is the efficiency of a single pair of roller bearings [11, 12].

The gear ratio of the drive moving the screens is

$$u = u_g u_{op} = 40 \times 6.545 = 261.8,$$

where $u_g = 40$ is the gear ratio of the reducing gear system; and $u_{op} = 6.545$ is the gear ratio of the open cylindrical gear system.

The power at the output shaft of the drive moving the screens is

$$P_{out} = P_{mo}\,\eta = 3 \times 0.78 = 2.34 \text{ kW},$$

where $P_{mo} = 3$ kW is the power of the MTKN 12-6 electric motor moving the screens.

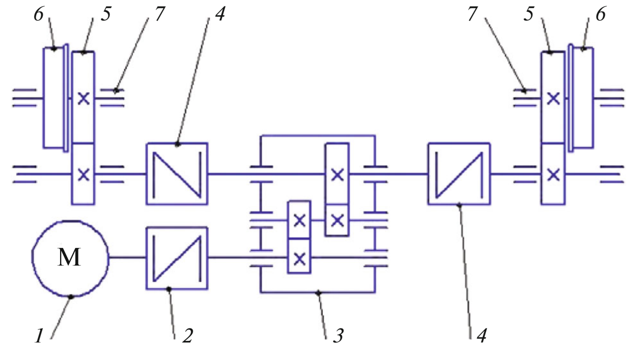

Fig. 4. Kinematic diagram of the drive for the mechanism moving the screens: (*1*) electric motor; (*2*) coupling; (*3*) cylindrical gear; (*4*) coupling; (*5*) open gear system; (*6*) drive wheels; (*7*) bearings.

The output shaft speed of the drive moving the screens is

$$n_{out} = n_{mo}/u = 905/268.1 = 3.46 \text{ rpm},$$

where $n_{mo} = 905$ rpm is the shaft speed of the MTKN 12-6 electric motor moving the screens.

Before modification, the torque at the shaft of the MTKN 12-6 electric motor moving the trolley is

$$T_{mo} = (P_{mo}/\omega_{mo}) \times 10^3 = (3/94.7) \times 10^3 = 31.7 \text{ N m},$$

where $\omega_{mo} = \pi n_{mo}/30 = (3.14 \times 905)/30 = 94.7$ rad/s is the shaft speed of the MTKN 12-6 electric motor moving the trolley.

The torque at the shaft of the drive wheel is

$$T_{ls} = T_{mo} u \eta_{co}^3\eta_{cl}^2\eta_{op}^2\eta_p^4 = 31.7 \times 261.8 \times 0.98^3 \times 0.97^2 \times 0.96^2 \times 0.99^4 = 6506.3 \text{ N m}.$$

To eliminate the design defects and improve screen operation, the drive moving the screens must be updated. The existing drive may expediently be replaced by a drive with improved kinematics and reliability, including a gear motor. That facilitates maintenance and repair of the screen, with corresponding decrease in the time, expense, and effort required [10]. A kinematic diagram of the updated drive moving the screens is shown in Fig. 5.

The overall efficiency of the updated drive moving the screens is

$$\eta = \eta_{co}^2\eta_{gm}\eta_p^2 = 0.98^2 \times 0.94 \times 0.99^2 = 0.88,$$

where $\eta_{co} = 0.98$ is the efficiency of the coupling [11, 12]; $\eta_{gm} = 0.94$ is the efficiency of the gear motor; and $\eta_p = 0.99$ is the efficiency of a single pair of roller bearings [11, 12].

For the mechanism moving the screens, we choose a conical–cylindrical gear system of K107R77DV100L4 type, with the following characteristics: motor power $P_{mo} = 3$ kW; shaft speed of electric motor $n_{mo} = 1383$ rpm; speed of gear motor's output shaft $n_{sl} = 3$ rpm.

The gear ratio of the drive is

$$u = n_{mo}/n_{sl} = 1383/3 = 461.$$

The torque at the shaft of the electric motor is

$$T_{mo} = (P_{mo}/\omega_{mo}) \times 10^3 = (3/144.8) \times 10^3 = 20.7 \text{ N m},$$

where $\omega_{mo} = \pi n_{mo}/30 = (3.14 \times 1383)/30 = 144.8$ rad/s is the speed of the electric motor's output shaft.

The torque at the output shaft of the gear motor is

$$T_{ls} = T_{mo} u_{gm} \eta_{co}^2 \eta_g = 20.7 \times 461 \times 0.98^2 \times 0.94 = 8614.9 \text{ N m}.$$

The speed of the drive wheel is

$$n_{dw} = \frac{1000 \times 60 v_{sc}}{\pi D_{dw}} = \frac{1000 \times 60 \times 0.06}{3.14 \times 400} = 2.9 \text{ rpm},$$

where $v_{sc} = 0.06$ m/s is the speed at which the screen moves; and $D_{dw} = 400$ mm is diameter of the drive wheel.

The calculated torque for the coupling is

$$T_{ca} = K_1 K_2 T_{sl} = 1.2 \times 1.0 \times 8614.9 = 10337.9 \text{ N m},$$

where the constant $K_1 = 1.2$ takes account of the contribution of the transmission; and the constant $K_2 = 1.0$ takes account of the coupling's working conditions.

We select an MZ-3-∅90/∅100 coupling with maximum torque $T_r = 8000$ N m at the maximum speed $n_{max} = 2800$ rpm.

A check calculation to prevent impermissible wear rates gives the result

$$p = \frac{T_{ca}}{0.9bd^2} = \frac{10337.9 \times 1000}{0.9 \times 30 \times 192^2} = 10.4 \text{ MPa},$$

where $b = 30$ mm is the tooth length; $d = m \times z = 4 \times 48 = 192$ mm is the diameter of the bush's splitting circumference; $m = 4$ is the engagement module, mm; and $z = 48$ is the number of teeth for the bush.

The permissible crumpling stress is $[p] = 12$–15 MPa. The result $p = 10.4$ MPa is less than this value and so the characteristics of the coupling are satisfactory.

Updating the screens simplifies their design and improves their reliability; increases the efficiency of the screen drives; reduces maintenance and repair costs and the frequency of unplanned downtime; and lowers production costs.

The additional capital costs for updating the screens amount to 3 146 160 rub, with recoupment over a period of 2.9 years. The product cost is reduced from 22 602.82 to 22 598.66 rub/t.

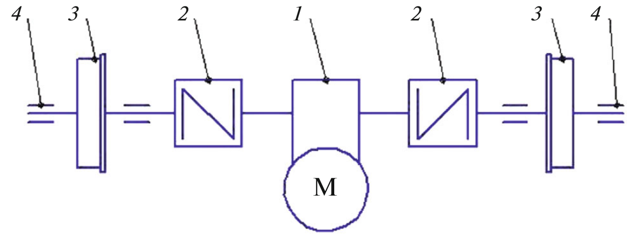

Fig. 5. Kinematic diagram of the updated drive for the mechanism moving the screens: (*1*) gear motor; (*2*) closed coupling; (*3*) drive wheels; (*4*) bearings.

CONCLUSIONS

1. In coke shop 1 at Ural Steel, double ten-roller screens are employed for primary sorting of coke, but they are subject to unplanned downtime on account of their poor reliability. Analysis of the drive designs for the coke-sorting stands and the mechanism for moving the screens indicates the following defects: large mass; complex design; difficulty in maintenance and repair; and low efficiency.

2. To improve their reliability, the existing drives of the sorting stands and the mechanisms for moving the screen are replaced by kinematically simpler and more reliable designs with a gear motor.

3. Updating the screens simplifies their design and improves their reliability; reduces maintenance and repair costs and the frequency of unplanned downtime; and lowers production costs.

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CONFLICT OF INTEREST

The authors of this work declare that they have no conflicts of interest.

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