DOI: 10.14529/power180113

CONSTRAINING THE DYNAMIC TORQUE OF A ROLLING MILL STAND DRIVE

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We herein consider issues of limiting the dynamic loads in the electromechanical systems of horizontal rolls in the stand of a plate mill. It is shown that shock loads mainly occur during gripping due to gaps in spindle connections; the size of the gap depends on how worn-out those spindles are. We analyze a method of limiting such dynamic loads that consists in gripping workpieces while accelerating the electric drive. We also task ourselves to prove the feasibility of implementing this method at the 5000 rolling mill employed by PJSC Magnitogorsk Iron and Steel Works (MMK). We present the oscillograms of workpiece gripping in accelerated and decelerated modes; those oscillograms prove how efficient pre-setting the gap size is. The paper contains a control chart that ensures electric drive acceleration before gripping. We have developed a mathematical model of the dual-mass electromechanical roll system for reversing stands and present herein the parameters of the simulated object. The proposed dynamic load limitation has been simulated. We herein specify a minimum timeframe between the onset of pre-acceleration and the load application moment, which timeframe is necessary for angular gaps to be fully closed. The timeframe value is explained. We have also implemented the developed speed control algorithms in the APCS of the 5000 rolling mill stand. The paper presents the torque and speed oscillograms for the workpiece gripping. It is proved that the torque overshoots in the upper- and lower-roll electric drives is reduced to acceptable levels by this method. We also analyze the oscillograms of the lower-roll drive parameters obtained over nine reversing-rolling passes. It is confirmed that the dynamic torque of grip does not exceed 25% of the steady-state torque. We thus conclude that this speed control method has proven efficient for the stand electric drives of the 5000 mill. Expanded implementation of this method in the existing hot-rolling mills is noted as practical.

Keywords: plate mill, reversing stand, automated electric drive, dynamic loads, limitation, control system, mathematical model, transient processes, experimental studies, pilot testing, implementation.

Introduction

Plate mill are crucial for making products for the further manufacture of large-diameter pipes by Russian metallurgy companies. Notably, over the last decade Russia has manufactured multiple rolling mills of this class, including the 5000 mill at MMK which was commissioned in 2009. This mill comes with a powerful reversing stand that increases the rolling force to 12 thousand tons.

The stand of this mill has individual roll drives based on synchronous M_1 and M_2 drives with frequency-adjusted speed [1]. Fig. 1 shows how the upper and the lower rolls gain their speed. The pass trajectory is generated by the APSC model based on the mill performance and rolling temperature criteria. The speed specification $V_{\text{active}}(t)$ is automatically transmitted from the first-level controller, which generates a table containing the desired trajectory points $S_{\text{active}}(t)$.

Acceleration and deceleration rates are mainly controlled by the interpolator. Its output sends the specific linear-speed signal $V(t)$ to the intensity setting device (ISD), which is imposes emergency restrictions on the job performance rate. The ISD output signal takes into account the roll diameter when specifying the angular speed of the drive, which specification is sent to the input of the closed speed loop.

A dynamic state characteristic of the reversing stand drives is the shock load it has to sustain when the rolls grip stock. This creates a torque overshoot resulting in dynamic speed drop. Literary review and experimental studies have shown the amplitude of such dynamic torque on hot-rolling mills might reach 50 to 100 percent of the steady-state torque [2–6]. This in its turn results in premature wear-out and breakages in mechanical equipment, especially in spindle heads. Splinde breakage is one of the worst emergencies, as it might result in the destruction of auxiliary equipment, causing long downtimes.

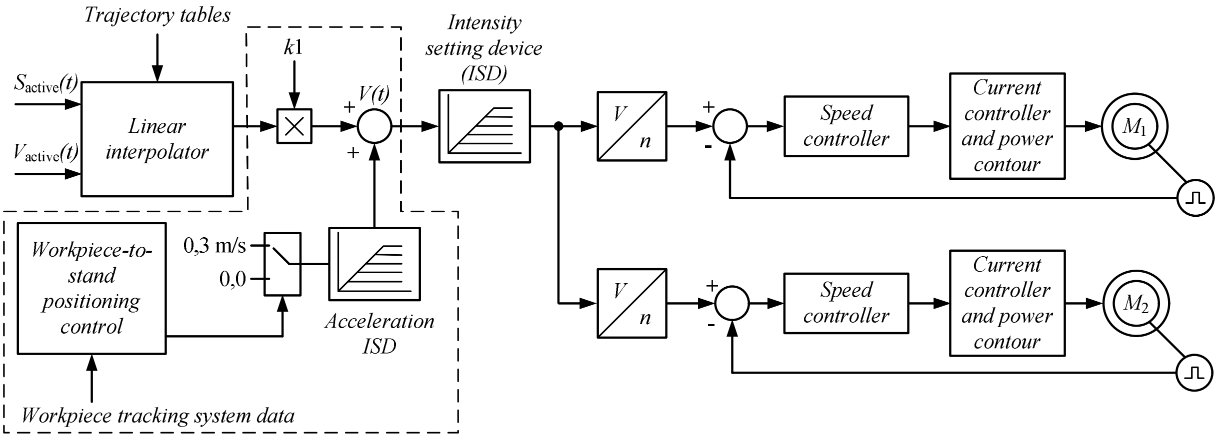


Fig. 1. Functional chart of the speed control system used by the stand drives (additional blocks of the implemented system outlined)

Statement of problem

One of the reasons behind dynamic loads emerging in the moment of gripping is that there are gaps in mechanical transmissions, the periodic opening and closure of which before capture causes the torque value to oscillate relative to zero. Closing the gaps after stock is in the stand causes additional shocks which results in elastic oscillations in the stand shaft lines.

As each roll of the 5000 mill has an individual drive without gears or pinion stands, dynamic shocks are caused by gaps in the spindle connections at each end of the transmitting shaft. The gap size depends on the spindle head wearout; it cannot be measured "on-line", but its effect on the dynamic state can be limited by the automated drive.

The basic technique aimed at reducing such gripping-caused shocks consists in pre-setting the gap size for the mechanical connections of shaft lines. The most popular method is to control such gripping by means of drive acceleration. This method is proposed in [7], has been studied by the Ferrous Metallurgy Institute of the Ukrainan National Academy of Sciences [8–10] and is now employed by multiple rolling mills, including the 1680 broad-strip hot-rolling mill used by Zaporozhstal [10, 11] and the 2500 hot-rolling mill used by MMK [12, 13]. Papers [14–16] present detailed studies on the transient processes taking place when stock is gripped. They have proved the method efficient for hot-rolling mills featuring geared drives and pinion stands.

Controlling the gripping-caused dynamic loads is a relevant problem for the 5000 mill. We should note that over its service life, it has experienced multiple spindle head breakages, which had severe consequences. This is why we tasked ourselves to find such control method that would limit dynamic loads, including the proposed method of pre-setting the gaps by means of drive acceleration for gripping. We had to solve the following problems:

1. Prove the feasibility of implementing this control method for our specified mill.

2. Choose the optimal acceleration value and time for full control of gaps in mechanical transmissions; prove those values feasible.

To solve these problems, we used a comprehensive approach including experimental studies and mathematical modeling. However, before we discuss our results it makes sense to describe the control system the algorithm whereof is used by the mill.

Main Part

Pre-acceleration Control Chart

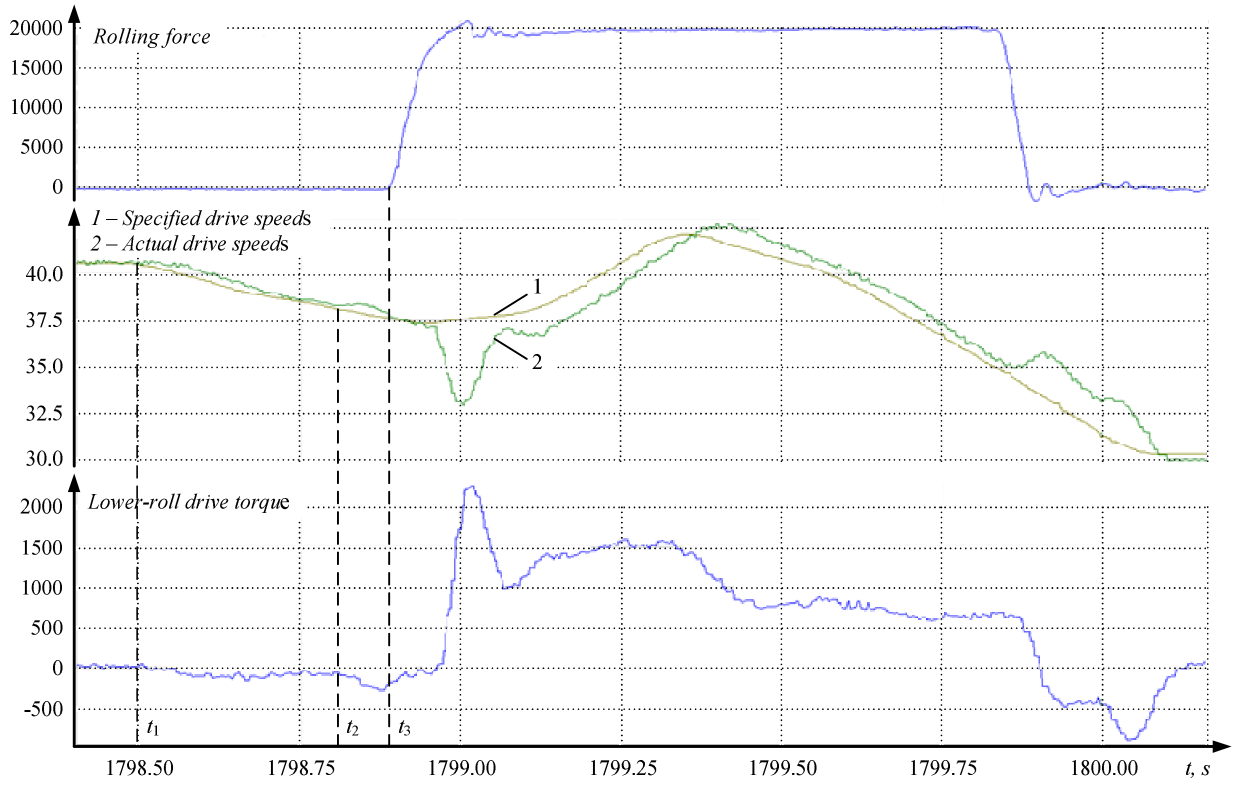
The simplified functional chart for the developed system that enables such pre-setting of gaps in mechanical transmissions was created by adding more functional blocks (outlined in Fig. 1) to the existing drive speed control chart.

To determine the pre-acceleration onset time, we use data from the existing roller-stock tracking system (the Workpiece Position Control block). To generate additional speed, we used an additional intensity-setting device (pre-acceleration ISD), the output of which is added to the interpolator output. Maximum linear speed is assumed to equal 0.3 m/s, whereas the additional acceleration rate is set experimentally at 0.22 m/s^2. These additional function are software-implemented in the stand controller.

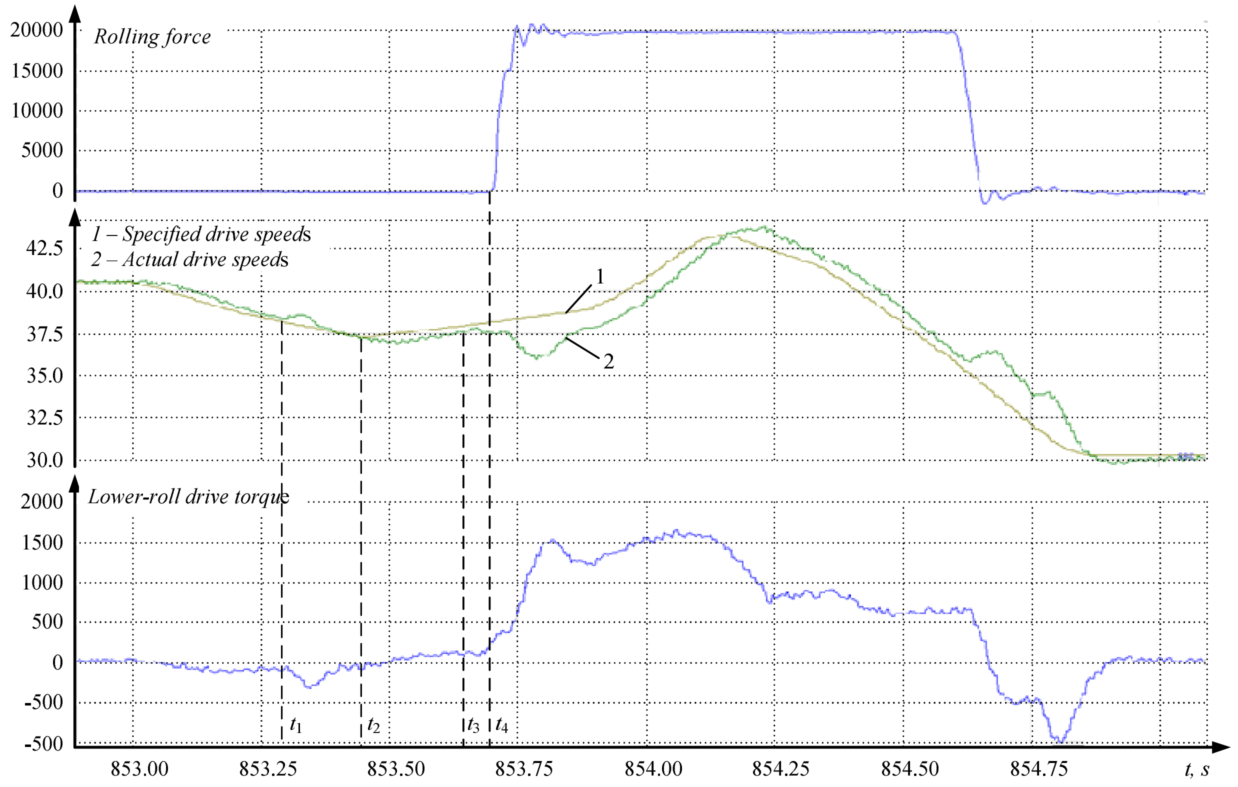
Proof of Concept: Implementing the method for the 5000 mill

To test the feasibility of implementing this method for the 5000 mill, we carried out an active experiment to compare the transient processes that occur at fully open and fully closed gaps in spindle connections when rolling identical profiles. For the first scenario, we gripped stock where the drives decelerated (see Fig. 2a for oscillograms); for the second scenario, we did the same where they accelerated, see Fig. 2b.

The first-scenario oscillogram shows a negative drive torque value in the deceleration section ($t_1 - t_3$). At t_2 it causes the gap to open, proven by a slight acceleration and a characteristic reduction in torque.



a)



b)

Fig. 2. Gripping oscillograms for fully open (a) and fully closed (b) gaps in spindle connections: window 1 is the rolling force, kN; window 2 is the specified and the actual drive speeds, rpm; window 3 is the lower-roll drive torque, kN·m

Table 1

Dynamic values of stock gripping

Figure no.	Speed drop			Torque overshoot		
	from	to	%	from	to	%
	rpm	rpm		kN·m	kN·m	
2a	37.5	32.6	15	1450	2200	51.7
2b	37.5	36	4.2	1450	1450	0

At t_3 stock is gripped with a fully open gap, causing undesirable speed and torque transients.

Fig. 2b highlights the timeframes which shed light on how gaps affect the transient process. At t_1, where the drive decelerates, the gap opens, which causes short-lasting deviations in torque and speed. Then the acceleration sign alters, and from t_2 to t_4 the drive accelerates, which is why the gap is closed at t_3, also shown by an insignificant speed deviation. At t_4 stock is gripped with the gap fully closed.

Table 1 gives a comparison of dynamic speed and torque values. If gripped with fully open gaps, a dynamic surge in torque occurs exceeding 50% of the steady-state value, whereas the dynamic speed drop amounts to 15%. In the second scenario, the dynamic torque does not exceed the steady-state value, whereas the dynamic speed drop is less than a third of its first-scenario value (only 4.2%).

The oscillograms and the table 1 data lead us to the conclusion that the torque overshoot was completely eliminated, whereas the dynamic speed drop was considerably reduced. This proves the feasibility of implementing this pre-acceleration method for the 5000 mill.

Dual-Mass Mechanical System Model

To solve the second problem, we developed a dynamic mathematical model of the electromechanical system of the 5000 mill roll [17]. As each roll has an individual electric drive, whereas mechanical connections do not contain any devices whose inertial could be on part with that of the drive and the adjusted inertia of the rolls, we assumed that the electromechanical system under analysis was a dual-mass system with elastic coupling and gaps in mechanical connections.

Fig. 3a contains the kinematic chart of the “drive and roll” system matching these conditions [18]. We use the following designations: $J_1 = J_M$ is the first-mass (drive mass) moment of inertia; J_2 is the second-mass (work and backup rolls, intermediate shafts, etc.) moment of inertia, adjusted; c_{12}, M_{12} rigidity and moment of the elastic coupling; β is the attenuation; M_1, ω_2 is the system entry torque and the system exit angular speed.

Fig. 3b presents the structural diagram of the model, where blocks 3, 5–7 are the typical blocks used in dual-mass system models. Block 4 determines the specifics of transient processes in the mechanical part, including the natural attenuation of oscillations, whereas block 5 simulates the gaps in mechanical connections. Speed feedback is simulated by block 8 with the factor k_{FS}.

Transfer function of the closed mechanical-art system (non-linearity caused by transmission gaps ignored),

$$W_{CM}(\mathrm{p}) = \frac{c_{12} \cdot W(p)}{c_{12} \cdot W(p) + J_1 \cdot p}.$$

To find that function, we carried out experimental studies that showed that the drive speed alterations caused by powering-off were generally attenuating oscillations. This is why the system had to be simulated by a second-order oscillatory link:

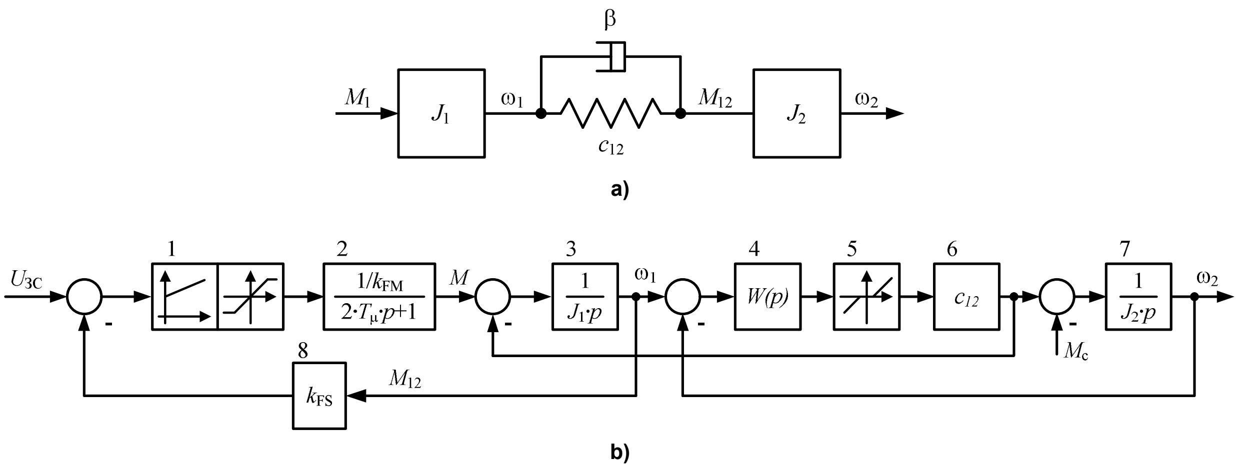


Fig. 3. Kinematic chart of the elastic mechanical transmission (a) and the structural diagram of the dual-mass electromechanical system (b)

$$W_{CM}(p)=\frac{1}{\mathrm{T}_1^2\cdot p^2+2\cdot T_1\cdot\xi\cdot p+1}.$$

Hence the block 4 transfer function

$$W(p)=\frac{\mathrm{J}_1}{\mathrm{T}_1^2\cdot c_{12}\cdot p+2\cdot T_1\cdot\xi\cdot c_{12}}.$$

The parameters for all links in the system were determined based on the equipment data and the rolling mill oscillograms. The model assumed that the torque control loop was set optimally. This is why it is represented as an aperiodic first-order link to simplify the analysis of processes in mechanical components (block 2). The speed control loop was set for a symmetric optimum with a proportionally-integral speed controller (block 1). As electric drive operations require torque restrictions, we've placed an adjustable-limit restriction block at its output.

Table 2

Source data for finding the dual-mass system parameters

Parameter	Value
Drive moment of inertia J_M	$125\,000\ \mathrm{kg\cdot m^2}$
Work roll mass G_{WR}	$63\,000\ \mathrm{kg}$
Work roll diameter D_{WR}	$1.2\ \mathrm{m}$
Backup roll mass G_{BR}	$226\,400\ \mathrm{kg}$
Backup roll diameter D_{BR}	$2.3\ \mathrm{m}$

Table 2 presents the parameters of the modeled object. It doesn't specify one important parameter, the spindle gap size that can be neither calculated nor measured experimentally. This is why the gap size values were taken from literatures and assumed to be from 1° to 10°; its effect is investigated further.

Simulating the dynamic torque control method

Simulation was done to analyze how the pre-acceleration value could affect the gap closing (e.g. cause a full or partial closure). The full closure of spindle gaps depends on two parameters: the pre-acceleration timeframe and the acceleration value. To ensure full closure at a specified acceleration rate, the time between the pre-acceleration onset and the rolls gripping the stock must be sufficient for spindle gaps to reach the desired value. As such, this time depends on the dual-loop speed control system settings as well as on the gap size.

Fig. 4 shows the graphs of the work-roll moment transients measured when simulating gripping with angular gaps of ±1°, ±5°, and ±10°. For source data, we assumed parameters obtained by simulating acceleration to the threading speed of 30 rpm (half the nominal speed) with subsequency pre-acceleration of 0.18 rad/s^2 or 0.22 m/s^2. Acceleration onset (initial graph point in Fig. 1) corresponds to time 3 s. The gap closure time is determined by the occurrence of dynamic torque on the work roll (in the model, 3b is the output of the block c_{12}).

We conclude from the graphs that with a ±1° gap, the time t_1 to full closure is 0.3 s; for ±5°, it is $t_2 = 0.7$ s; for ±10°, $t_3 = 1.75$ s. Before those intervals are over, gaps are not fully closed, which is why applying any load is highly undesirable. Besides, one has to take into account the torque transient process which must be over before stock is gripped.

Fig. 5 shows the temporal dependency of drive torque and speed when gripping stock while accelerated, at the specified transmission gap sizes. We conclude that a ±10° gap and an acceleration of 0.22 m/s^2 the provided pre-acceleration time did not suffice for full gap closure. This causes a 10.7% speed drop

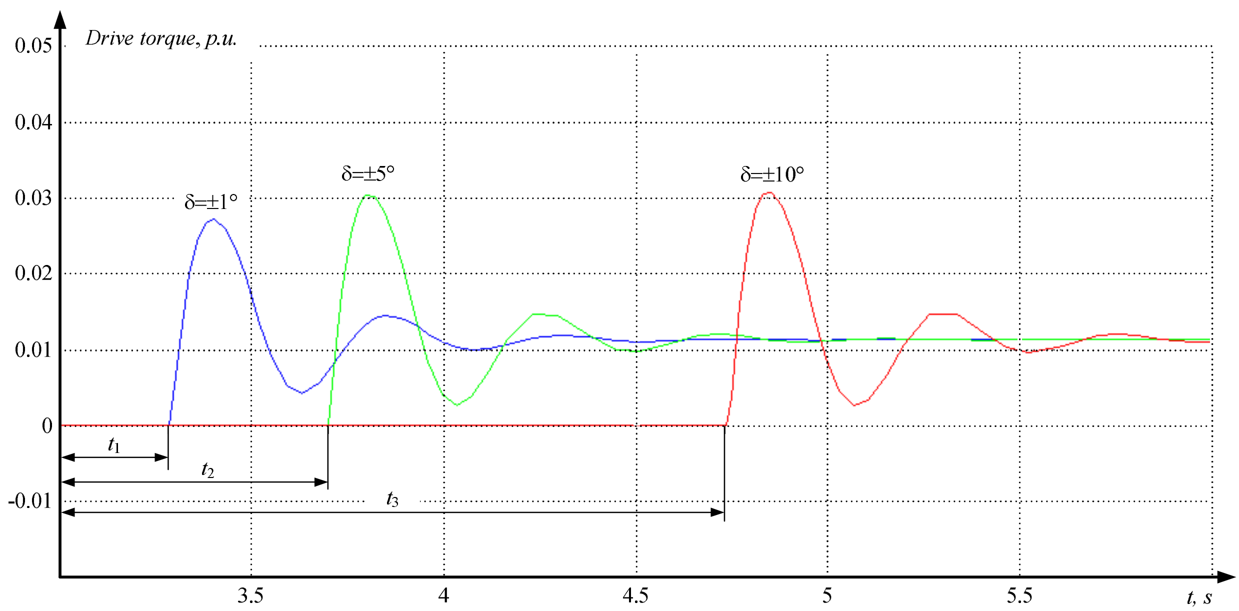


Fig. 4. Transient processes of the work roll torque for various angular gaps (p.u.)

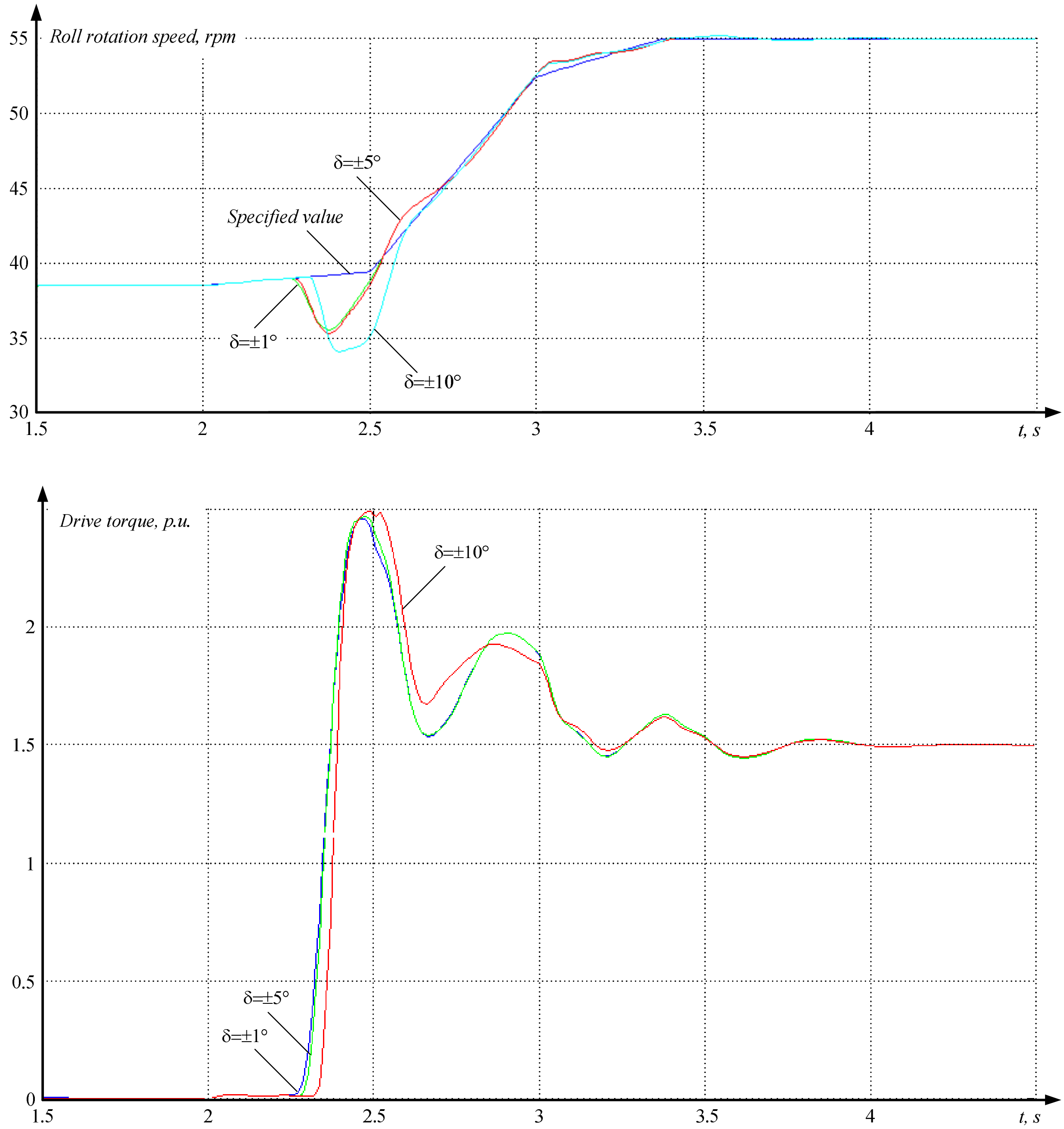


Fig. 5. Electric drive speed and torque at gripping

(from 37.5 to 33.5 rpm), whereas it does not exceed 4.5% in case of full gap closure. As a result, this increases the dynamic torque peak, which thereafter engages the speed controller output block. As acceleration measurements require intervention with the speed control system settings, it makes sense to prolong such pre-acceleration so that the gap is fully closed.

Simulation showed that if acceleration lasted longer than 0.5 seconds, the graphs of the speed and torque transient processes coincided no matter the gap size.

Therefore, picking optimal pre-acceleration time for a specified acceleration has a considerable impact on the transient processes in the electric drive coordinates. To that end, we set a 0.5-second timeframe between the acceleration onset and the load application, which guaranteed full closure of gaps of up to ±10°.

Experimental studies

Fig. 6a shows oscillograms of processes that occur when stock is gripped with the proposed control algorithm using the acceleration and the pre-acceleration time values as found optimal above. Fig. 6b shows a small section of the same oscillograms, zoomed-in. What differs it from similar oscillograms in Fig. 2 is that drive torques are given in percent of the nominal torque value. Note that steady-state values in the range of 180 to 200 percent of the nominal value

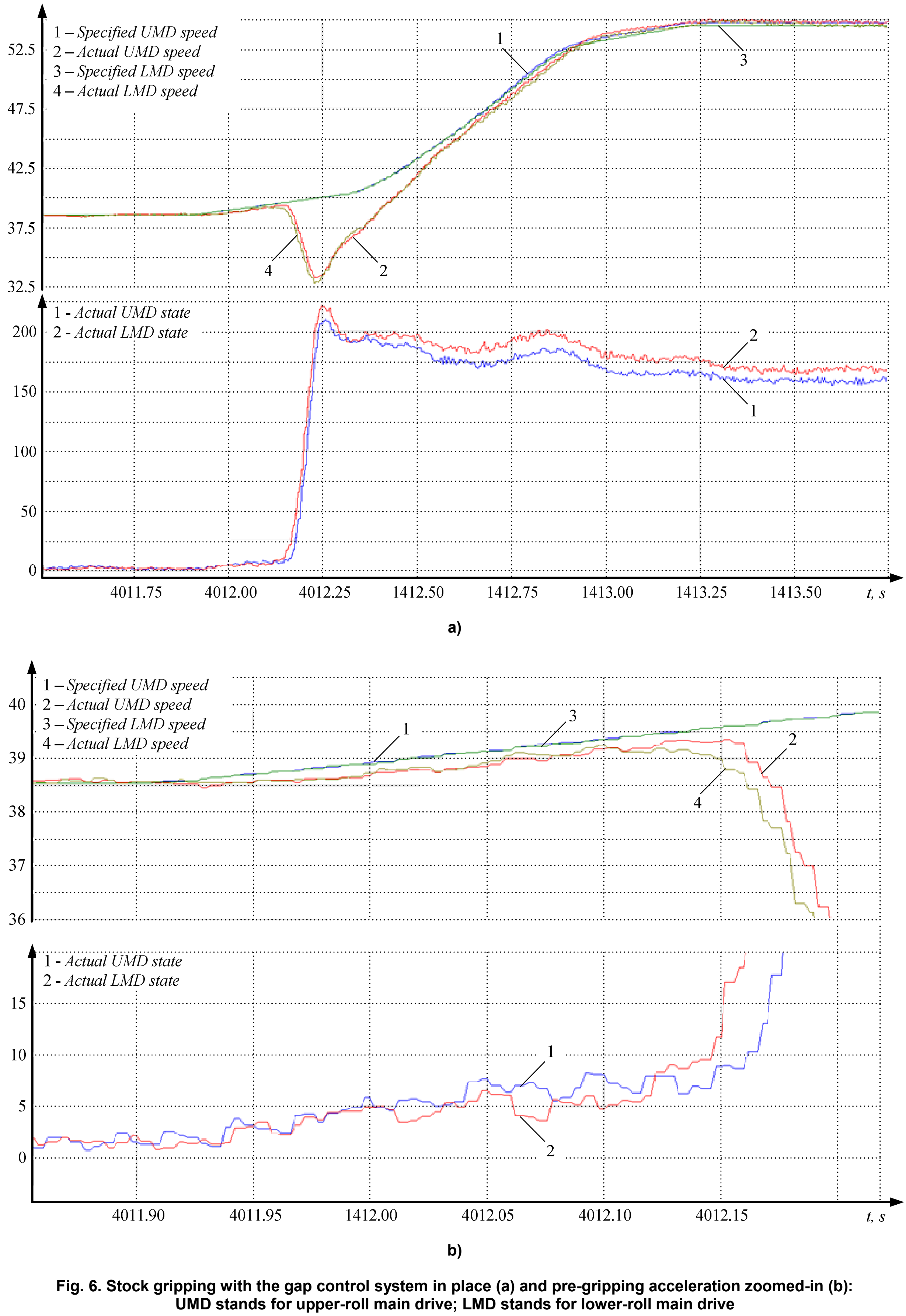


Fig. 6. Stock gripping with the gap control system in place (a) and pre-gripping acceleration zoomed-in (b): UMD stands for upper-roll main drive; LMD stands for lower-roll main drive

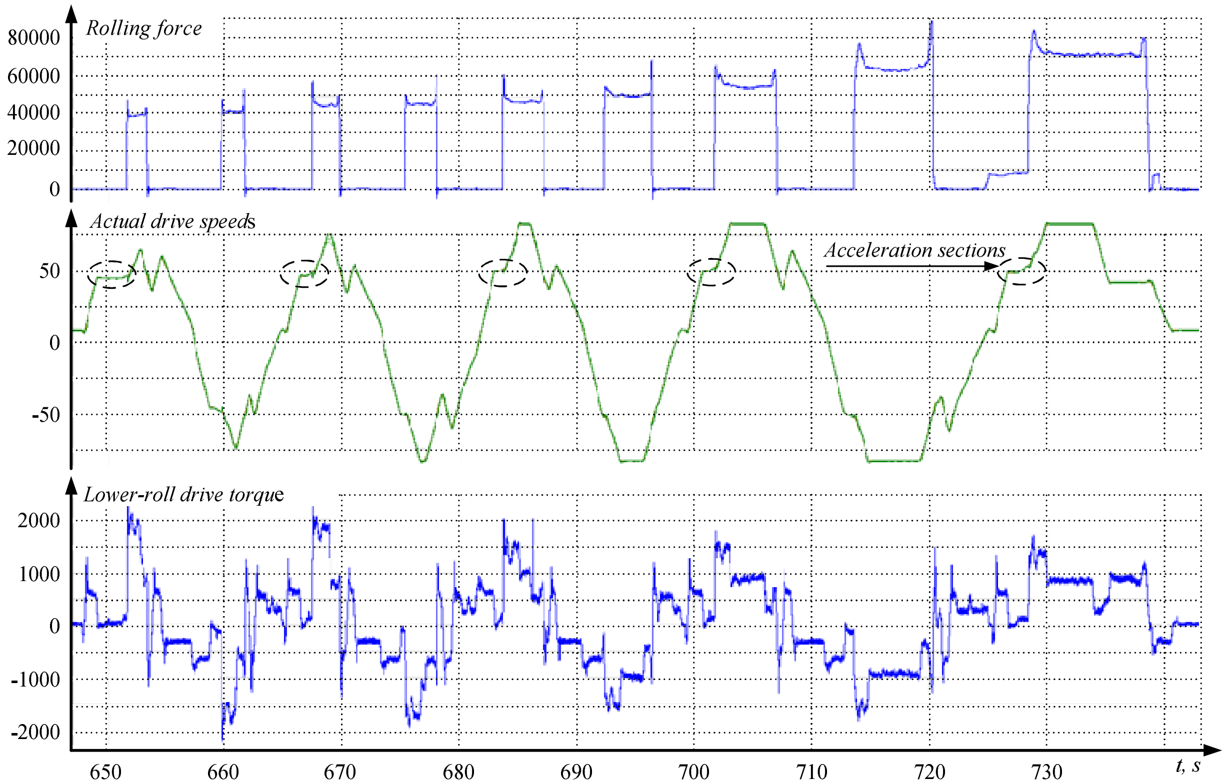


Fig. 7. Oscillograms of the lower-roll electric drive parameters for the finishing

are acceptable for reversing stand drives, which is why they are characteristic of the many specimen of the 5000 mill range. Dynamic loads occurring at gripping reach up to 240% of the nominal torque value, which is the limit. It is obvious that such overloads are not acceptable.

Before stock is gripped starting at ~4011.925 s, the specified speed is increasing gradually, which creates a positive dynamic torque shown in Fig. 6b. This is why the torques of both drives are positive before gripping with no alternating-sign torque. This is what ensures the preliminary closure of gaps. For LMD, the torque overshoot is 26.4%, which is acceptable. UMD shows a similar reduction in torque. Therefore, these oscillograms proves the method efficient.

Generalization of the results

The proposed electric drive control method was implemented in the APCS controller algorithm for all reversing-rolling passes. This is whown by oscillograms obtained in nine finishing rolling passes, see Fig. 7. That pre-acceleration was achieved is shown by the pre-gripping “shelf” in the speed oscillograms (before rolling force gains in the first window).

Oscillographic analysis leads us to a conclusion that the dynamic torque of gripping does not exceed 25% of the steady-state value in any of the passes. Torque overshoots only occur in the first three passes, whereas the dynamic increase in the torque is not significant in later passes. This proves the full closure of spindle connection gaps that was achieved while sticking to the recommended acceleration and pre-acceleration duration values.

In general, our experiments have proven the feasibility of implementing the proposed reversing-stand drive speed control algorithms for the 5000 mill.

Conclusions

Literature review and experimental studies have shown that an angular gap in the spindle connections of the shaft lines of a plate mill stand results in unacceptable dynamic loads to be sustained by electromechanical systems when the rolls grip stock. We have herein presented a method for controlling the drives so that such spindle-connection gaps are pre-set by roll pre-acceleration.

We have developed a dynamic mathematical model of the dual-mass electromechanical system found in a horizontal stand. Pre-setting of gaps has been simulated. As a result, pre-acceleration time and acceleration intensity have been found to significantly affect the transient processes of drive coordinates. We have determined the optimal ratios of these parameters.

An algorithm has been designed that provides acceleration-enabled gripping, which algorithm has further been added to the ASCP of the 5000 mill used by MMK. Experimental studies of transient processes occurring in a single rolling cycle have confirmed torque overshoot being reduced to 25% and the speed drop being reduced to 4.5% of their respective steady-state values.

Industrial-scale implementation of this solution can reduce dynamic loads sustained by the electrical and mechanical equipment of horizontal stands. The research results are thus recommended for implementation in the roughing 2000 mill group at MMK as well as in other rolling mills whose drives have to sustain shock loads.

Research supported by the Russian President's Grant МД-979.2017.8

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Received 20 February 2018

УДК 621.311 + 621.34.001 DOI: 10.14529/power180113

ОГРАНИЧЕНИЕ ДИНАМИЧЕСКОГО МОМЕНТА ЭЛЕКТРОПРИВОДА КЛЕТИ ПРОКАТНОГО СТАНА

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Рассматриваются вопросы ограничения динамических нагрузок в электромеханических системах горизонтальных валков клети толстолистового прокатного стана. Показано, что одной из основных причин возникновения ударных нагрузок при захвате является наличие зазора в шпиндельных соединениях, величина которого зависит от степени их износа. Рассматривается способ ограничения динамических нагрузок за счет обеспечения захвата заготовки в режиме ускорения электропривода. Поставлены задачи обоснования целесообразности внедрения данного способа на стане 5000 ПАО «Магнитогорский металлургический комбинат» (ПАО «ММК»). Представлены осциллограммы захвата заготовки в режимах ускорения и замедления электропривода, подтвердившие эффективность предварительного выбора зазоров. Приведена схема системы управления, обеспечивающая ускорение электроприводов перед захватом. Разработана математическая модель двухмассовой электромеханической системы валков реверсивной клети, представлены параметры моделируемого объекта. Выполнено моделирование рассматриваемого способа ограничения динамических нагрузок. Обоснован минимальный временной интервал между началом предварительного ускорения и моментом приложения нагрузки, необходимый для полного закрытия угловых зазоров. Выполнено внедрение разработанных алгоритмов управления скоростными режимами в АСУ ТП клети стана 5000. Представлены осциллограммы момента и скорости, зафиксированные при захвате заготовки. Доказано, что перерегулирования моментов электроприводов верхнего и нижнего валков при реализации предложенного способа снижаются до допустимого уровня. Рассмотрены осциллограммы параметров электропривода нижнего валка, полученные за девять проходов реверсивной прокатки. Подтверждено, что динамический момент при захвате не превышает 25 % установившегося момента. Сделан вывод о технической эффективности внедрения предложенного способа управления скоростными режимами в электроприводах клети стана 5000. Отмечена целесообразность расширенного внедрения на действующих станах горячей прокатки.

Ключевые слова: толстолистовой прокатный стан, реверсивная клеть, автоматизированный электропривод, динамические нагрузки, ограничение, система управления, математическая модель, переходные процессы, экспериментальные исследования, опытно-промышленная эксплуатация, внедрение.

Работа выполняется при поддержке гранта Президента РФ МД-979.2017.8

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Поступила в редакцию 20 февраля 2018 г.

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Constraining the Dynamic Torque of a Rolling Mill Stand Drive / V.R. Khramshin, V.R. Gasiyarov, A.S. Karandaev et al. // Вестник ЮУрГУ. Серия «Энергетика». – 2018. – Т. 18, № 1. – С. 101–111. DOI: 10.14529/power180113

FOR CITATION

Khramshin V.R., Gasiyarov V.R., Karandaev A.S., Baskov S.N., Loginov B.M. Constraining the Dynamic Torque of a Rolling Mill Stand Drive. *Bulletin of the South Ural State University. Ser. Power Engineering*, 2018, vol. 18, no. 1, pp. 101–111. DOI: 10.14529/power180113