11th International Research/Expert Conference
”Trends in the Development of Machinery and Associated Technology”
TMT 2007, Hammamet, Tunisia, 05-09 September 2007.

ANALYSIS OF DYNAMIC WORKLOAD CAUSED BY IMPACTS IN THE DRIVE ELEMENT GAPS ON THE LINE ROLLING TRACKS

Doc.dr. Mustafa Imamović
Prof.dr. Ahmet Hadžipašić
Mechanical Faculty Zenica, B&H

ABSTRACT
Determination of proper workloads in the rolling stands drive elements is a very complex task. This complexity is caused by the technological process specifications which are conditioned to a large number of influential factors which are stochastic in nature and application. Determination of quantity as well as impact of the dynamical loads caused by drive elements' gaps are separate issue. This paper covers the analysis of the rolling process on line rolling tracks, way these dynamic workloads are caused by impact in the drive elements' gaps as well as proposed solution for reduction of the impact of the gap sizes to the dynamic workloads. This procedure enables reduction of the total drive elements' workloads which would for sure lead to the safer and more reliable facility operation.
Keywords: dynamic workload, rolling track, drive element gaps, impacts

1. INTRODUCTION
Under the influence of external forces (motor momentum, breaking force, technology resistances and friction) elements in the drive lines are getting deformed making the elements gain not only basic rotation movement but angle oscillations as well. As a result, drive elements are dynamically burdened with additional burdening forces (momentums and elasticity forces) which are undergoing changes through time momentum along with the speed of its own oscillatory system.
Besides, rolling stand drive element sets are constructed and fabricated with significant sizes of clearances (toothed coupling, toothed carrier, hinged cylinders…). In the period of non-stationary operation (startup, acceleration and breaking, stopping), due to existing clearances (gaps), impacts are significant, additional dynamic burdening forces which may reach significant values.
This paper shows an option of implementation of physical model as well as an analysis of dynamic impact burdens in the elements of lined rolling mills (tracks).

2. ROLLING PROCESS ON LINED ROLLING MILL

2.1. Rolling mills classification

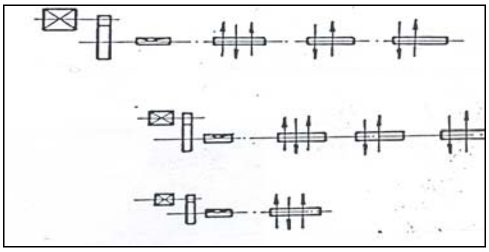

Figure 1. Stand layout in rolling mills

Depending on the purpose and capacity we can encounter rolling mills with different layout and number of rolling stand. Rolling mills are often being named according to the number and layout of the rolling stands: single stand, lined, tandem, staggered, etc.[1] Single stand mills have only one rolling stand. If more stands are placed into single line, this mill is called opened or lined mill, Figure 1. Number of stands in one lined mill can vary. Mills are driven either by one motor on one side or both sides.
Rolls in all stands have identical RPM, therefore a rolling piece in the end of the section, has the same speed when elongated just as when it was short, which is a serious flaw in lined rolling mills.

Rolling mills with two or more lines have been built to remedy this flaw. The simplest case is the mill with two stands placed on from the other, so called tandem mill.
Material is being rolled in few passes, first on the first stand and later on the second one. Each stand is driven with separate motor. This type of mills is most used for thick tin plate rolling.

2.2. Physical model of drive line

Rolling stands drive lines consist of a number of elements with different stacks, randomly positioned along the drive line. Aiming easier and more accurate results of the calculations, realistic drive lines are replaced with appropriate equivalent elastic kinetic-model / physical model, which provide accuracy for determined sizes and figures. [2]
Argumentative pass from realistic drive line to equivalent elastic-kinetic model and accuracy of determining the model parameters is essentially influencing the accuracy of the dynamic calculation. An example of equivalent elastic-kinetic model with two rotating forces is given in Figure 2, where:

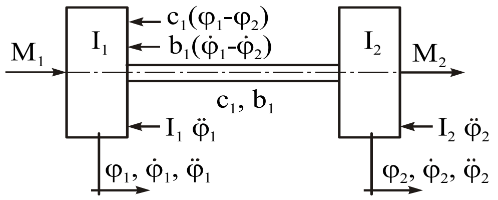

Figure 2. Equivalent elastic-kinetic model with two rotation forces

I_1, I_2 – reduced inertia momentums
M_1 – electro motor drive momentum
M_2 – external resistance momentum
c – equivalent link rigidness
φ_1, φ_2 – angled (generated) movement coordinates
b_1 – attenuation coefficient

In realistic conditions, drive lines have relatively big number of forces / weights: electro motor rotor forces, gearbox element forces with couplings, transfer stand, hinged cylinder and rolling rolls. Therefore, physical model of the rolling mill drive line usually has three or more concentrated (reduced) number of forces. Bigger input of forces in the model gives more accurate results but it also gives bigger number of differential equations and more complex calculation.

3. DYNAMIC IMPACT BURDENS IN THE DRIVE LINES GAPS

Rolling stand drive lines sets are constructed and fabricated with a significant gap sizes. In the period of non-stationary operation, gaps are causing impacts, additional dynamic burdens which may reach significant values.

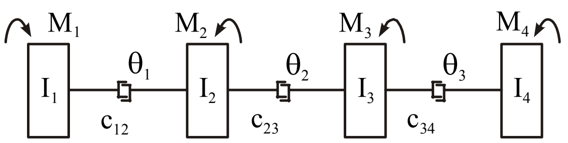

Figure 3. Four-force rotational system with gaps

Marks on Figure 3. are:
I_1, I_2, I_3, I_4 - reduced force inertia momentum
M_1 – motor momentum / drive momentum
M_2, M_3, M_4, - external momentum
θ_1, θ_2, θ_3, θ_4 – radial gaps in the elements 1÷4

Movement of this system can be observed in multiple stages and these observations enable dynamic burdening forces analysis due to element gaps.

3.1. Stage one - electro motor rotor startup

Differential equation which describes motor rotor movement in the startup stage can be presented as:

$$I_1 \frac{d^2\varphi_1}{dt^2} = M_1 \qquad \dots (1)$$

Taking into account initial conditions, for $t = 0$, equation for electro motor rotor startup for stage one has this form:

$$\varphi_1 = \frac{M_1}{2I_1} t^2 \qquad \dots (2)$$

3.2. Stage two

After the gaps are closed θ_1 two-forced system I_1, I_2 continues movement where differential movement equation of two-forced system has this form:

$$I_1 \frac{d^2\varphi_1}{dt^2} + c_{12}(\varphi_1 - \varphi_2) = M_1 \qquad I_2 \frac{d^2\varphi_2}{dt^2} - c_{12}(\varphi_1 - \varphi_2) = -M_2 \qquad \ldots(3)$$

Solution for differential equations (3) can be defined in the following form:

$$\varphi_1 = A\cos pt + B\sin pt + \frac{M_a}{I_1 p^2} \qquad \varphi_2 = -\frac{I_1}{I_2}\left(A\cos pt + B\sin pt + \frac{M_a}{I_1 p^2}\right) \qquad \ldots(4)$$

Elastic forces momentum M_{12} in the impact moment is determined by the equation:

$$M_{12} = M_a(1 - \cos pt) + \frac{\omega_0 c_{12}}{p}\sin pt \qquad \ldots(5)$$

First part of the expression (5) is elastic forces momentum from the effect of external burden caused by the gap in the elements between forces 1 and 2. It does not depend on the size of the gap θ_1, while the other directly depends on θ_1. Dynamic impact momentum in the gaps can be defined as:

$$M_3 = \frac{\omega_0 c_{12}}{p}\sin pt \quad \text{or} \quad M_3 = \sqrt{2M_1\theta_1 c_{12}\left(\frac{I_2}{I_1 - I_2}\right)}\ \sin \mathrm{pt} \qquad \ldots(6)$$

Amplitude of the additional dynamic burdens caused by the elastic impact in the gaps is increased depending on the size of gaps as per parabolic function. For two-force systems with gap, dynamic coefficient during the external burden effect can be more then 2, and equation used to describe it can be presented as:

$$k_d = 1 + \sqrt{1 + \left(\frac{\omega_0 c_{12}}{M_a p}\right)^2} = 1 + \sqrt{1 + \left(\frac{2M_1\ n_2\ c_{12}}{M_a^2}\right)\theta_1} \qquad \ldots(7)$$

3.3 Stage three

After the gap closing θ_2 all three forces are continuing movement defined by the differential equations:

$$I_1 \frac{d^2\varphi_1}{dt_2^2} + c_{12}(\varphi_1 - \varphi_2) = M_1 \qquad I_2 \frac{d^2\varphi_2}{dt_2^2} - c_{12}(\varphi_1 - \varphi_2) + c_{23}(\varphi_2 - \varphi_3) = -M_2$$

$$I_3 \frac{d^2\varphi_3}{dt_2^2} - c_{23}(\varphi_2 - \varphi_3) = -M_3 \qquad \ldots(8)$$

3.4. Stage four – continued forces movement

After the gap closing θ_3, all four forces continue their movement which can be defined by differential equations [2].

Equations for defining momentum M_I, M_{II} and M_{III}:

$$\begin{aligned} M_I &= c_{12}\, a_{12(1)}\ A_1 + c_{12}\, a_{12(2)}\ A_2 + c_{12}\, a_{12(3)}\ A_3 \\ M_{II} &= c_{23}\, a_{23(1)}\ A_1 + c_{23}\, a_{23(2)}\ A_2 + c_{23}\, a_{23(3)}\ A_3 \\ 0 &= c_{34}\, a_{34(1)}\ A_1 + c_{34}\, a_{34(2)}\ A_2 + c_{34}\, a_{34(3)}\ A_3 \end{aligned} \qquad \ldots(9)$$

Coefficients A_1, A_2, and A_3 are determined by the equations (9) and these are used to determine momentums M_{12}, M_{23} and M_{34} in the elastic links after closing of the gaps θ_1, θ_2 *and* θ_3.

4. IMPACT OF GAPS ON DYNAMIC BURDENS

Aiming the exact impact of gaps on dynamic burdens of the rolling mills drive elements, experimental research is required. These researches should be executed for different operational conditions to provide a number of results and therefore reach relevant conclusions.

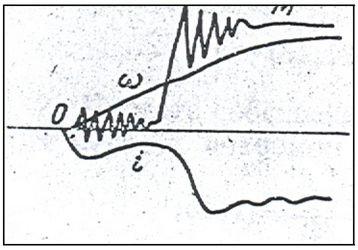

Figure 4. Impact burden during grabbing of material

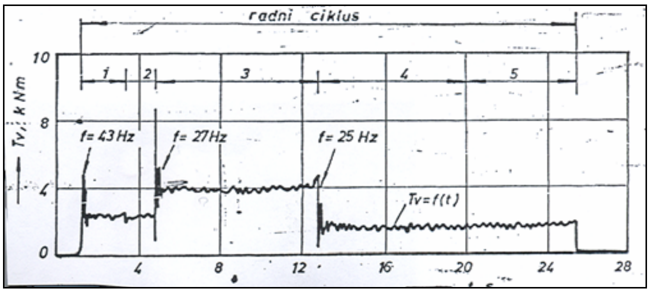

Figure 5. Momentum flow during rolling

Figure 4 gives change flow of angle speed ω, motor momentum M and motor power during material grabbing on lined rolling mill rolling stand. Parallel to this, Figure 5 shows momentum flow change during multiple passes in the lined rolling mill. These two figures, which are results of experimental research, show significant change of the size of dynamic burden on the elements in the moments when material is grabbed between two rolls, as shown in theoretical analysis in this paper. In this process, using the real momentum flow data, we can define ration between dynamic and static momentum as factor of dynamic load K_d . ($K_d = M_d\ max / M_v$) [3] .

Factor K_d is different for different rolling mills, and the difference is present in various locations on a single rolling mill. Therefore, it is necessary to carefully exercise the analysis for determining appropriate burdens and the way of inputting of K_d into rolling mill element calculations.

Reducing the impact of gaps on dynamic burdens is possible in more ways: inputting elastic elements, good fabrication of toothed carriers and rolling mill acceleration control.

5. CONCLUSION

According to the analysis given in this paper, certain conclusions concerning dynamic burdens caused by rolling mills gaps, which are:

- Additional dynamic burdens in the rolling mill elements are caused by gaps.
- Incorporating the physical model of rolling mills enables analytical analysis of dynamic burdens caused by gaps.
- Experimental research of burden flow is required to provide exact impact of gaps on dynamic burdens.
- Experimental measurement results have shown existence of different impact factors in different positions on the same rolling mill as a result of forces layout and different gaps.

6. REFERENCES

[1] Čaušević M.: Rolling metal processing, Veselin Masleša, Sarajevo 1983

[2] Иванченко Ф., Полухин В. -.: Динамика и прочность прокатиюго оборудования, Металлургия, Москва, 1987.

[3] Imamović M: „Constructional choice aspects and reliability calculation of basic parts of rolling stands“, Doctorate research, Zenica 2004.