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THE RAILWAY VEHICLES OF „SERBIAN RAILWAYS“ WITH WHEEL SLIP CONTROL

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Abstract: *In this paper, the electro locomotives made by the license of ASEA, Sweden, in the Companies Koncar, Croatia, and Electroputere, Romania with the traction electromotors for wavy direct current has been simulated with effective wheel slip control strategies in MATLAB Simulink environment. A converter fed traction electromotors for wavy direct current has been used for this analysis. Besides, a mathematical model of traction load model has been developed. This model was then integrated with the traction electromotors for wavy direct current with effective wheel slip control. The performance analysis of the drive for various mechanical conditions between wheel and rails has been made.*
Key words: *wheel slip control, traction load, electro locomotive.*

1. INTRODUCTION

The electro locomotives made by the license of ASEA, Sweden, in the Companies Koncar, Croatia, and Electroputere, Romania, are in service for more than forty years. Right from the beginning problems with fast wearing of driven gear and axle fractures occurred. These two occurrences represented “the now generation of problems” for the Railway Authorities in the Balkans which have been using both types of locomotives.

The propulsion system of the electro locomotives made by the license of ASEA is a mechanical system which comprises the traction electromotor for wavy direct current (3), a cogged clamp (2), a cardan shaft (5), a rubber clamp, a reductor (1), the driving shaft (4) and a monoblock wheel (Fig. 1) [1].

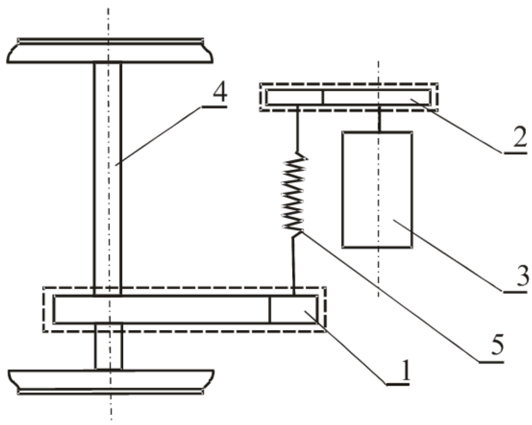

Figure 1. *Propulsion system of the electro locomotives made by the license of ASEA*

This mechanical system with dc controllable drive enabled the use of high power 25 kV, 50 Hz or 15 kV $16\frac{2}{3}$ Hz supply in electric traction.

With closed loop dc drive, it became possible to control acceleration of the motor by controlling the motor torque. This was done by controlling the voltage from a low value at start to the high value at high speed. Thus the motor torque can be controlled such that it accelerates with high value acceleration.

However, a high acceleration of the dc motor alone does not guarantee a high acceleration of the train which also depends upon the wheel- rail interaction.

Thus, the rotation of the wheel (a consequence of motor speed) may not completely result in translational motion of the train. When this happens, the wheel starts slipping on the track. This results in fast wearing of driven gear (1) and generating a strong torsion oscillation viz. fracture of the driving shaft (4)[1].

Consequently, the authors of this paper has given a mathematical formulation of a valuable insight into wheel-rail interaction of the electro locomotives made by the license of ASEA. In this paper, an attempt has been made to simulate the traction load and apply it to a dc drive with effective slip control strategies. The converter configuration used is two stage sequence control of two half controlled converters connected in series. This converter configuration has advantages of better power factor and lower current ripples.

2. MATHEMATICAL MODELLING OF TRACTION LOAD

The mechanics of traction deals with the nature of traction load and the dynamics of wheel-rail interaction. This includes slip- spin effects of the wheel on the track. These effects are governed by the complex interaction between traction motor, its controller and wheel-rail adhesion characteristics. This interaction is crucial to the motion of the locomotive but it can also introduce certain characteristics which are undesirable, such as wheel slip/spin.

The motion of the locomotive is induced by the force created when the wheel pushes against stationary track. The torque applied by the motor is transferred as a force at the rim of the wheel. This force at the rim of the wheel is transferred to a force which propels the train through a mechanism known as adhesion. Adhesion may be defined as the ability of the wheel to exert the maximum tractive effort without sliding on the rail. This adhesion force is the virtue of friction between the two surfaces (the wheel and the rail). Under the condition of acceleration there is a dynamic unbalance in the adhesion force and force exerted by the traction motor. This causes slipping motion of the train on the track. Friction causes the wheel to catch up and prevents the slipping motion. However, the surface could not possibly have such an effect on the wheel once the wheel has achieved pure rolling motion and constant angular and linear velocity. Thus friction becomes zero when the train moves on a flat track at constant velocity. Under pure rolling the train moves under the effect of rolling friction.

Rolling friction is caused primarily by the interference of small indentations formed as one surface rolls over another. This is because the wheel and the surface will undergo deformations due to their particular elastic characteristics. At the contact points, the wheel flattens out while a small trench is formed in the surface of the rail. The normal force is now distributed over the actual contact area rather than the point just below the center of the wheel. The load seen by the traction motor is also the result of adhesion as the force transferred by the adhesion is also seen as the load on the traction motor.

Adhesion is the ability of the wheel to exert the maximum tractive effort on the rail and still maintain the persistence of the contact without sliding. Thus the coefficient of adhesion can be approximately defined as the ratio of maximum tractive effort that can be developed without slipping and the weight on the driving axle. Thus, the force of adhesion is given by:

$$(1)\quad F_a = \mu \cdot N$$

where μ is the coefficient of adhesion and N is the normal reaction force at the point of contact.

The coefficient of adhesion is affected by a number of factors. Contaminants like oil, grease, water, snow and mud reduces its value. Its value also depends upon the torque speed characteristic of the traction motor. The traction motor with lower speed regulation has higher value of coefficient of adhesion. The value of coefficient of adhesion decreases slowly with wheel slip after initial sharp rise. The measure of wheel slip is provided by a factor called wheel slip ratio which is given by:

$$(2)\quad \lambda = \frac{\Delta v}{v_w} = \frac{v_w - v_t}{v_w}$$

where v_w is the peripheral velocity of wheel (referred as wheel velocity) and v_t is the velocity of train.

A change in slip ratio causes change in coefficient of adhesion. This is shown in Figure 2. Besides, the coefficient of adhesion decreases with wheel speed.

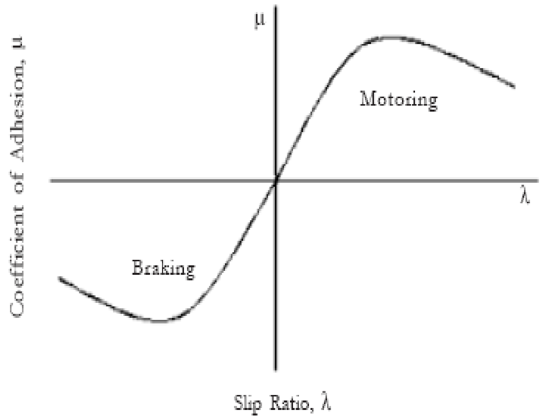

Figure 2. *Variation of coefficient of adhesion with wheel slip ratio for various track conditions*

The traction load comprises of the various frictional forces that act on the system and the gravitational pull that act if the train is moving on an inclined track. The various frictional forces that act on the traction system can be classified as internal, external and air friction [2] .

Internal friction comprises of the friction at the bearings, guides etc. External friction comprises of the rolling friction between the wheels and the rails, and friction between wheel flanges and rail. This depends upon the nature of the track. Air friction is independent of the weight of the train but depends upon the shape and the size of the train, velocity and direction of wind and speed of the train.

All these components collectively form the train resistance. The train resistance can also be identified in terms of common classification of friction such as windage, viscous friction, coulomb friction and stiction which has a very large value. However, owing to large value of the inertia, particularly due to the weight of the vehicle, accelerating torque forms the major component of total torque in the accelerating range.

The overall dynamics of the train can be broadly classified as:

- wheel dynamics; and
- train dynamics.

The differential equation defining the wheel dynamics is given as:

$$(3)\ J_0 \frac{d\omega_0}{dt} = M_0 - M_{ot} - F_a \cdot \frac{D}{2}$$

where J_0 – the total inertial moment (the sum of inertial moment of the larger gear of the jagged reductor -180 Nms^2; inertial moment of the driving shaft -340 Nms^2 and inertial moment of the monoblock wheel -1600 Nms^2). The total inertial moment is $J_0 = 2120\,Nms^2$; ω_0 – angular speediness of the monoblock wheel; $F_a = \mu \cdot N$ - the force of adhesion; N - the normal reaction force at the point of contact (20 *daN*); D – diameter of the monoblock wheel (D=1210 mm); M_{ot} - transient value of load torque contributed by friction (external friction), windage and gravitational component and is given by

$$(4)\ M_{ot} = F_{ot} \cdot \frac{D}{2} = (F_{ext} + \varepsilon \cdot M_g \cdot \sin\theta) \cdot \frac{D}{2}$$

where ε is the minimum distance between the radius of the wheel and the horizontal line passing through the centre of gravity; F_{ext} - the external friction.

Angular speediness of the shaft of the monoblock wheel (ω_0) and transient value of motor torque (M_o) are

$$(5)\ \omega_0 = \frac{\omega}{i}$$

$$(6)\ M_0 = \eta \cdot i \cdot M_m$$

where ω – angular speediness of the shaft of the traction electromotor for wavy direct current; i- transfer ratio of the jagged reductor ($i = 3{,}65$); $\eta = 0{,}975$ (grade of utility according to the IEC- 349).

The dynamic balance of the shaft of the traction electromotor is described by the next equation:

$$(7)\ J_m \frac{d\omega}{dt} = M - M_m$$

where J_m is the inertial moment of rotating mass with the angular speediness ω. The inertial moment J_m is a sum of inertial moment of the traction electromotor for wavy direct current (550 Nms^2), inertial moment of the cogged clamp (2 Nms^2), inertial moment of the cardan shaft (3 Nms^2), inertial moment of the rubber clamp (10 Nms^2) and inertial moment of the lesser gear of the jagged reductor (10 Nms^2). Therefore, the inertial moment is J_m= 575 Nms^2 [1]; $M(t)$ – transient value of torque at the shaft of the traction electromotor for wavy direct current; $M_m(t)$ - transient value of torque oncoming from idler force.

Equation (3) can be further solved as

$$(8)\ \omega_0 = \int \frac{M_0 - M_{0t} - F_a \dfrac{D}{2}}{J_0} \cdot dt$$

The angular velocity of the wheel yields the peripheral speed of the wheel which will be addressed as wheel speed.

$$(9)\ v_w = \omega_0 \cdot \frac{D}{2}$$

The train dynamics governs the translational motion of the train (vehicle). This can be given by a first order equation in terms of mass, train resistance and tractive effort. This is given as he equation of the mechanical system running is:

$$(10)\ m\frac{dv}{dt} = F_v - F_{ot}$$

where: m – mass of each wheel set (20 daN); F_{ot} - total reaction forces defined with equation (4). This equation can be solved as

$$(11)\ v_T = \int \frac{F_v - F_{ot}}{m} \cdot dt = \int \frac{F_v - F_{ext} - \varepsilon \cdot M_g \cdot \sin\theta}{m} \cdot dt$$

The torsion moment of the driving shaft (4) (Fig. 1):

$$(12)\ M_t = k \cdot \Delta\theta$$

$$(13)\ \Delta\theta = \theta_0 - \theta$$

$$(14)\ v_w = \frac{d\theta_0}{dt} \qquad \text{in the complex domain}$$

$$\theta_0 = \frac{v_w}{s}$$

$$(15)\ v_T = \frac{d\theta}{dt} \qquad \text{in the complex domain}$$

$$\theta = \frac{v_T}{s}$$

where k – torsion constant of driving shaft (4) (Fig. 1). The torsion constant of the leaves part of the driving shaft (i.e. part of the driving shaft from the jagged reductor to the near monoblock wheel) is $k_1 = 553\,10^6\ Nm \cdot rad^{-1}$. Torsion constant of the longer part of the driving shaft (i.e. part of the driving shaft from the jagged reductor to the further monoblock wheel) is $k_2 = 9{,}8 \cdot 10^6\ Nm \cdot rad^{-1}$ [1]; θ_0 - banking of driving shaft induced of motor torque; θ - banking of driving shaft induced of load torque.

Thus, the mechanical load model is represented by Figure 3.

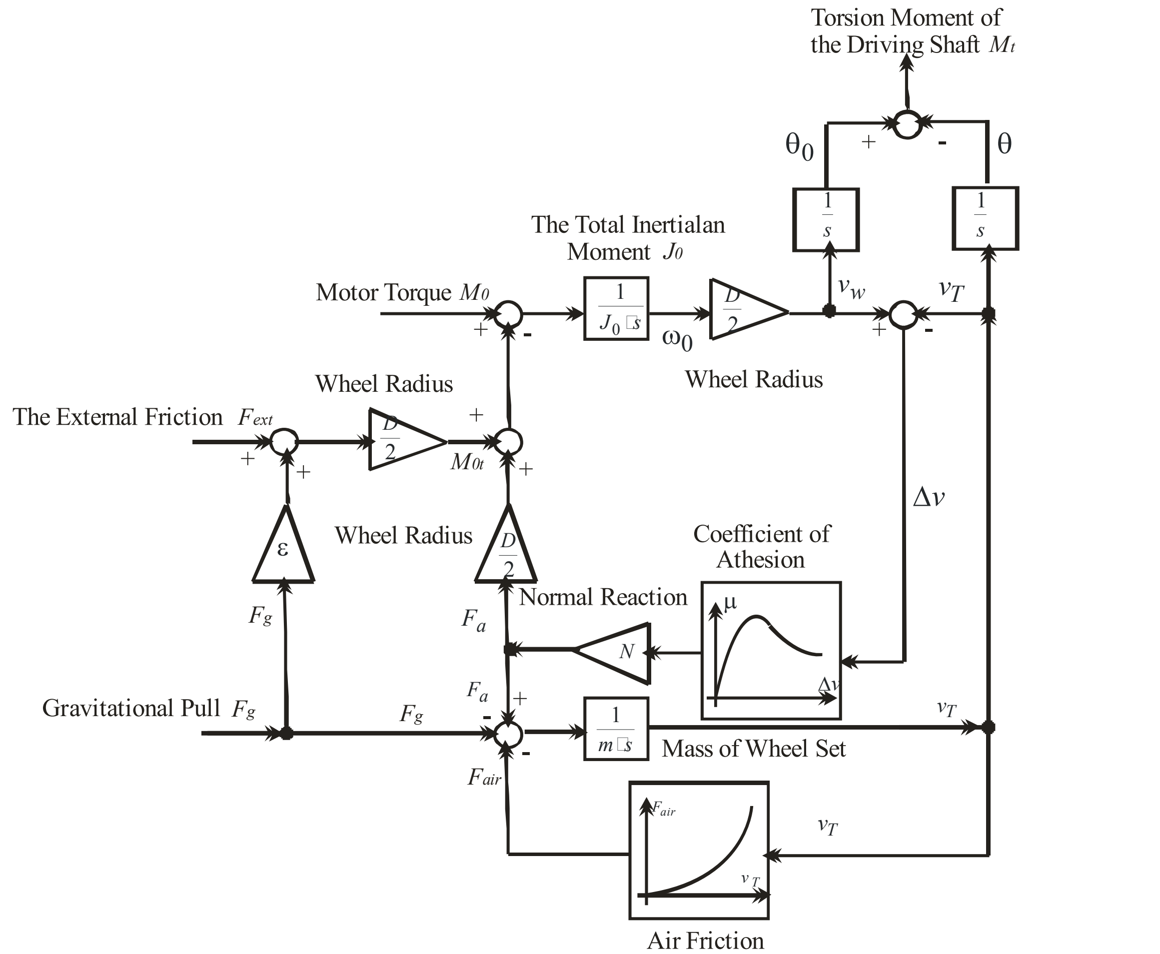

Figure 3 *Block diagram for entire mechanical load system*

3. TORSION MOMENT AT A SLIPPAGE OF THE WHEEL SET

Based on the former equations, we made a program in MATLAB-SIMULINK to simulate the torsion moment at the wheel set of the electro locomotives made by the license of ASEA, Sweden, in the Companies Koncar, Croatia, and Electroputere, Romania. We received a chronological variety of the torsion moment of the longer part of the driving shaft according to Fig. 4 when we started from this simulation program. We assumed that a slippage of the wheel set appeared because of a nuisance value of the traction coefficient at $M_{ot*} = \dfrac{M_{ot}}{M_{0n}} = 1$

(The traction coefficient at this environment is defined by the following term: $F_v > \mu \cdot m \Rightarrow \mu < \dfrac{M_{0n}}{\dfrac{D}{2} \cdot Q_a} = \dfrac{27924{,}8849}{\dfrac{1{,}21}{2} \cdot 200000} = 0{,}23$).

We also assumed that the rotating moment of the shaft of the traction electromotor for wavy direct current of the electro locomotives made by the license of ASEA is determined by [1]:

$$M(t) = \frac{33}{32} \cdot k_0 \cdot I_{sr}^2 (1 + \frac{16}{33} \cos 2\omega t + \frac{1}{33} \cos 4\omega t) = M_{sr} (1 + a_1 \cos 2\omega t + a_2 \cos 4\omega t)$$

where M_{sr} - in between value of the torque of the shaft of the traction electromotor for wavy direct current; $a_1 = \frac{16}{33}$ - factor amplitude of a frequency $2f = 2 \cdot 50 = 100 Hz$; $a_2 = \frac{1}{33}$ - factor amplitude of a frequency $4f = 4 \cdot 50 = 200 Hz$.

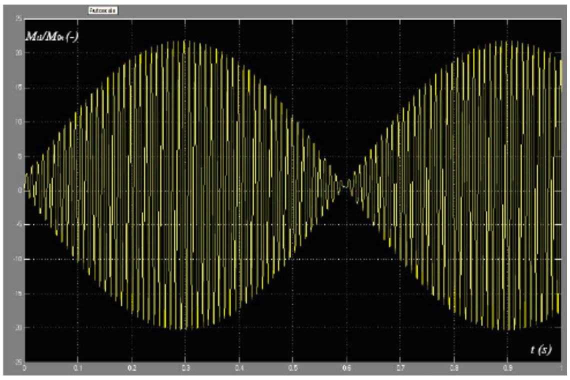

Fig. 4: *Comparative value of the torsion moment of the longer part of the driving shaft at* $a_1 = \frac{16}{33}$ *of a frequency* $2f = 2 \cdot 50 = 100 Hz$ *during the slippage of the wheel set*

Based on Fig. 4, we can conclude that the torsion moment of the longer part of the driving shaft quite quickly rises during the slippage of the wheel set. This moment achieved the value of $\frac{M_{t1}}{M_{0n}} = 23$ ($M_{t1} = 6{,}42$ MNm) in a quite short period of $t \le 0{,}3 s$. Consequently, the torsion moment during the slippage of the wheel set will permanently impair the longer part of the driving shaft. This may results in fast wearing of driven gear and generating crevices and fractures in the wheel set.

4. WHEEL SLIP CONTROL

It may be observed from Figure 5 that the entire slip range is divided into two regions, namely, stable and unstable region. In the stable region of operation the value of coefficient of adhesion increaseswith slip. In the stable region, as the motor torque increases, the wheel speed increases, resulting in immediate increase in the value of slip, which causes increase in force of adhesion. This results in increase of the mechanical load on the system causing wheel-slip to improve. Thus there is a self regulating action.

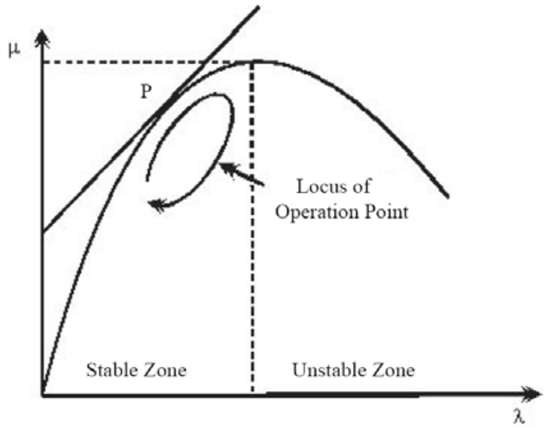

Figure 5 *The* $\mu - \lambda$ *curve showing stable and unstable regions and gradient at operating point*

However, in the unstable region, coefficient of adhesion decreases with wheel-slip. In the unstable region, as the motor torque increases, the wheel-slip increases causing reduction in force of adhesion. This results in decrease of mechanical load on the system causing further acceleration of wheel. The slip ratio increases further. This is a cumulative action resulting in slipping of the wheel on the track.

The goal of all slip control methods is to control the slip in order to prevent wear of the wheels and the rail and to use the present adhesion effectively. Optimizing methods adds a search of the maximum adhesive force. This is achieved when the slip is controlled towards the peak of the slip curve within the stable region as shown in Figure 5. The two slip control strategies suggested towards this end are

- Slip control to the reference value; and
- Gradient control method.

In slip control to the reference value, the slip error is processed by a slip ratio controller (a PI controller) which produces a torque command for the motor. Thus, the motor accelerates at a reference slip under the stable region such that the adhesion force is optimally utilized. Once the train reaches the desired speed, the speed controller takes control and maintains the speed at desired value. To obtain an optimal reference slip, it is sufficient to know the $\mu-\lambda$ characteristic of the track. Once, this characteristic is known, the optimal slip can be estimated by a choosing a value of slip which is near the peak of the $\mu-\lambda$ curve.

However, this method has a drawback that when the track condition changes the reference slip may no longer be optimal and the it may happen that for the same operating slip, the train operates in the
unstable zone of $\mu-\lambda$ characteristic. To overcome this disadvantage, gradient control scheme is adopted.

4.1 Gradient control scheme and simulation results

In this scheme, the gradient of $\mu-\lambda$ curve for the track is calculated and the operating point is controlled by a PI controller, such that, the gradient $\left(\frac{d\mu}{d\lambda}\right)$ at operating point is equal to the reference value. The idea of the scheme is to make the train operate at an operating point sufficiently high on the μ−l curve, such that, the force of adhesion is optimally utilized without moving into the unstable zone of operation even when the track condition is changed.

It may be noted in Figure 5 that the gradient of $\mu-\lambda$ curve is positive in the stable region and unstable in the negative region. Further, as one moves towards the peak of the curve, the slope continuously decreases and at the peak of the curve it becomes zero. Thus one can ensure an optimal utilization of adhesion force by selecting a small positive value of gradient as a reference value. A block diagram representation of a gradient control scheme is given in Figure 6.

The gradient detector detects the slope of the $\mu-\lambda$ curve at the point of operation. This slope is then made to follow up the reference gradient with the help of gradient controller.

Figures 7(a)-7(c) shows the response of the drive when subjected to different track conditions. The train moves first 35 m on track 1. The track condition changes immediately at this point and the train moves on track 2 for next 45 m. It again shifts to track 1 after this.

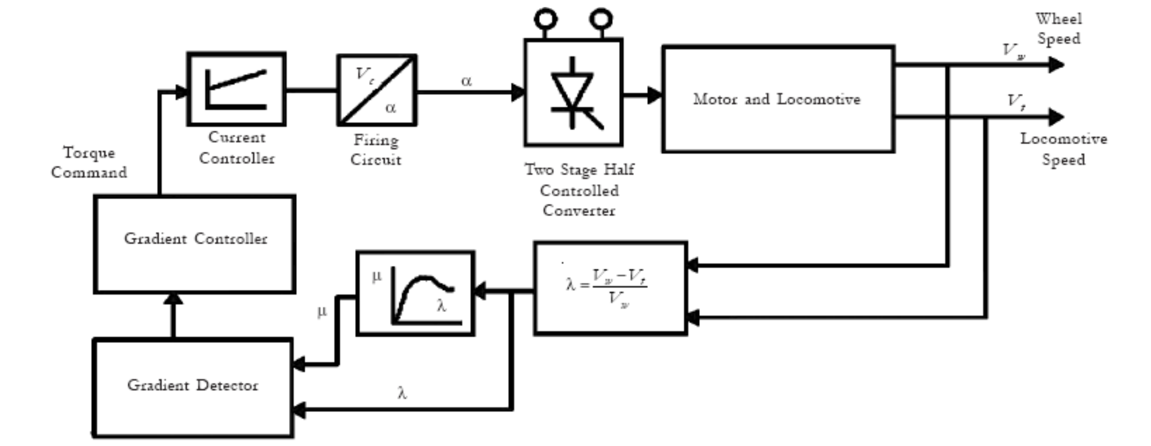

Figure 6 *Block diagram of gradient control scheme*

It is evident from Figure 7(a) that the motor torque has been controlled such that the gradient of $\mu - \lambda$ curve is commanded to the reference value even when the track condition is changed. Figure 7(b) shows the locus of the $\mu - \lambda$ curve followed by the train. It reveals that the train repeatedly moved from stable to unstable zone. This was the result of the oscillation due to the tuning of controller. However, soon after, the train settled at an optimum slip governed by the reference value of gradient. This is given by point a in Figure 7(b). When the track condition changed the point of operation immediately shifted from a to *a'*. Then, after a small transient, the train settled at optimum slip for track 2. This is shown by point *b'*. Now, when the track condition is changed back to the track 1, operating point of the train drops back to point b and starts building up towards the optimum slip for track 1. Figure 7(c) shows the build up of train and wheel speed towards the desired speed.

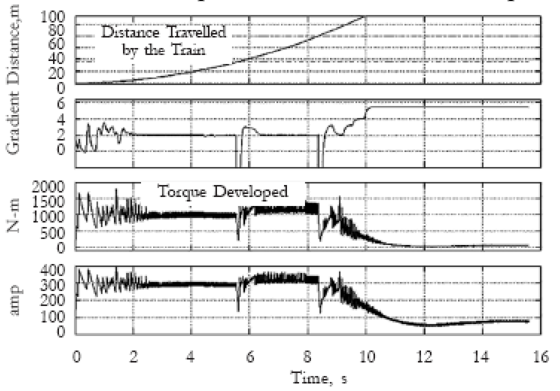

Figure 7(a) *The response of the system showing corresponding armature current and torque developed by the motor as the slip ratio is being controlled through gradient control to the commanded value while the train builds up to the desired value of speed on a variable track. (Simulation for the variable track: Parameter specifications: Reference gradient= 2°; Desired speed= 80 km/h; Nature of track = changing track).*

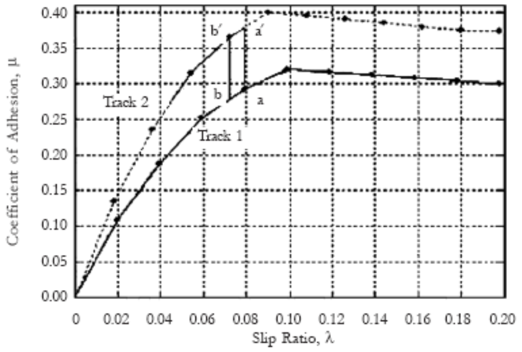

Figure 7(b) *Locus of the* $\mu - \lambda$ *curve followed by the train*

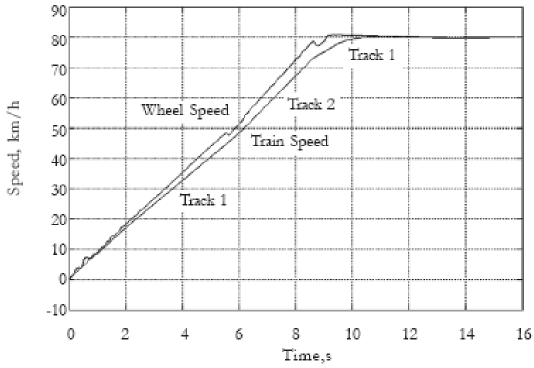

Figure 7(c) *Variation of wheel speed and train speed with time*

5. CONCLUSIONS

An important study made in this work was the modelling of traction load and its heavy dependence on wheel-rail interaction. The electric drive of the electro locomotives made by the license of ASEA, Sweden, in the Companies Koncar, Croatia, and Electroputere, Romania is integrated with the traction load model which provides extremely useful insight into the nature of traction load and pattern of tractive effort demand which forms the basis of the requirements from an electric traction drive. The simulation results clearly indicate the traction drive perceives a very heavy load torque during the acceleration which reduces to a small value once the acceleration period is over and the train has resumed pure rolling motion. Consequently, the torsion moment during the slippage of the wheel set will permanently impair the longer part of the driving shaft. This may results in fast wearing of driven gear and generating crevices and fractures in the wheel set.

The traction drive of the electro locomotives made by the license of ASEA, Sweden, in the Companies Koncar, Croatia, and Electroputere, Romania is modeled to provide wheel slip control with a concept of gradient control to demonstrate how the wheel slip ratio can be controlled to make optimal use of force of adhesion between train wheel and the track so as to allow maximum acceleration. The model prepared here can be used to provide a simulation study of electric traction drive performance employing wheel slip control under various conditions such as motion of the train on a flat track, motion on an inclined track and motion of the train on a changing track.

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ЖЕЛЕЗОПЪТНИТЕ ВОЗИЛА НА “СРЪБСКИТЕ ЖЕЛЕЗНИЦИ” С КОНТРОЛ НА ПЛЪЗГАНЕТО НА КОЛЕЛАТА

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Ключови думи: *контрол на плъзгане на колелото, тягово натоварване, електролокомотив*
***Анотация**: Настоящата статия представя симулацията в MATLAB Simulink среда, чрез ефективни стратегии на контрол, на плъзгането на колелата на електролокомотивите с тягови електрически двигатели на изправен ток, произведени по лиценз на ASEA, Швеция в Koncar, Хърватия и Electroputere, Румъния. За целите на този анализ са използвани конверторно захранвани тягови електродвигатели на изправен ток. Освен това е разработен математически модел за тягово натоварване. Този модел е интегриран впоследствие с тяговите електродвигатели на изправен ток с ефективен контрол на плъзгане. Осъществен е анализ на задвижването в разнообразни механични условия между колелото и релсите.*