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ANALYSIS OF DYNAMIC MOMENTS OF TORSIONAL DRIVE OF MACHINERY*

Ahmed S. Alhunaishi+
Mechanical Eng. Department, Faculty of Engineering, University of Aden.

ABSTRACT

The capital of dynamic behaviour of drives has been the theme of articles of technical literature. Due to the increasing use of metal industry machines, it is necessary to expand calculation methods to enable the complex dynamic analysis of torsional driving systems of such machines.

The dynamic facts of the drive may be classified to tasks of analysis and tasks of synthesis. In dynamic analysis task, one is mainly interested in controlling and checking the most important parameters in an existing system or in a proposed system. Analysis enables estimation of the stress in individual parts .Also, one can estimate important data, such as the position of critical values of characteristics, maximum values of deformation of flexible elements etc. .

In the task of synthesis, one is interested in the basic data and methods of solution for the choice of mass ratios of stiffness, respecting also the choice of family of loads, proper choice of clearances etc., to fulfill the assumption of the admitable load and the motion of proposed system.

Among the metal industry machines the most important are the press mill machines. In the process of using these machines there is a row of problems, some of which are solved and some other are waiting for a solution. One of those solved is the analysis of the drive of the press mill and the working machine with regard to the characteristics (moments) of the drive (motor).

KEY WORDS: Torsional Drive, Moments Of Drive, Dynamic Behavior Of Drive, Mathematical Modal Method.

ANALYSE DES MOMENTS DE TORSION DYNAMIQUES ENTRAÎNEMENT DES MACHINES *

RÉSUMÉ

La capitale du comportement dynamique des entraînements a été le thème d'articles de la littérature technique . En raison de l'utilisation croissante des métaux machines de l'industrie , si est nécessaire d'étendre les méthodes de calcul afin de permettre l'analyse dynamique complexe des systèmes de conduite de torsion de ces machines .

Les faits dynamique de l'entraînement peuvent être classés à des tâches d'analyse et les tâches de synthèse . Dans l'analyse dynamique tâche on est surtout intéressé dans le contrôle et la vérification des paramètres les plus importants dans un système existant ou dans un système proposé . L'analyse permet une estimation de la contrainte dans les parties individuelles. Aussi, on peut estimer les données importantes, telles que la position des valeurs critiques des caractéristiques , des valeurs maximales de déformation d'éléments flexibles , etc .

Dans la tâche de synthèse , On est intéressé dans les données de base et les méthodes de solution pour le choix des rapports de masse de rigidité , en respectant aussi le choix de la famille de charges , bon choix de jeux , etc , Pour répondre à l'hypothèse de la charge admitable et le mouvement du système proposé .

Parmi les machines de l'industrie du métal le plus important sont les machines de presse de l'usine. Dans le processus d'utilisation de ces machines là-bas une rangée de problèmes , dont certains sont résolus et d'autres sont en attente d' une solution . L'un d'eux est retrouvé à l'analyse de l'entraînement du broyeur de la presse et la machine de travail en ce qui concerne les caractéristiques (moments) de l'entraînement (moteurs) .

MOTS CLÉS: torsion d'entraînement , des moments de Drive, comportement dynamique de l'entraînement , mathématique Modal méthode

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+ Contact author (mddwh@yahoo.com)

1. INTRODUCTION

Looking after cheap designs of minimum mass and cost of machine often lead to unfavourable dynamic properties of the machines. This situation influences the increase of vibrations and noise during operations which has the decisive influence on service life and operation safety [14].

In recent years, premature failure of machine drive components has been experienced for reasons not explained by the conventional method of stress analysis used in machine design. This has focused attention on the problem of transient torsional vibration in machine drive systems. The machine designers and machine users have developed a technology to predict reliably the dynamic behaviour of the rolling machines. The objective achieved is the development of a reliable mathematical model, combining the dynamics of the electrical and mechanical systems along with a computer simulation to establish the transient and steady-state behaviour of rolling machines [25].
The drive system of a rolling machine consists of two subsystems: mechanical spring - mass system, and electrical drive and control system. The dynamic behaviour of the drive system is largely influenced by the interaction of the two subsystems. Consequently, mathematical models must take into account the dynamic characteristics of each component of the subsystems [10].

The interaction between the electrical and mechanical systems is the importance conclusions. The dynamic characteristics of the mechanical system must be established at the design stage and must be compatible with the response characteristics of the electrical system. All possible sources of disturbance must be identified and their effect for the various modes of vibration for the entire system must be analyzed [11].

The rolling torque disturbance is a function of time. This torque build up is a function of the reduction, the roll diameter and the speed of rolling at the time the workpiece enters the

machine. The motor torque in a speed regulated system is affected by any speed change in the system. The amount of speed drop during material entry and overspeed as the material leaves the machine is dependent on the dynamic characteristics of the complete drive system and the response characteristics of the electric components of the control system. The sufficient large vibration amplitudes of the mechanical system cause premature drive system failure [21].

Metal industry machinery is subjected to very severe operating conditions due to intentional loading for maximum production output and exposure to high dynamic loading from large moving masses being processed [9].
As a result of torque variation on a drive system, fatigue is generally the criterion of failure. In addition, due to the large size of the drive system components, brittle fracture is also common. For fatigue and fracture mechanics analyses to yield reliable results, precise stress levels must be used. Spindles and couplings are geometrically complex components and require advanced methods of analysis for determining fairly accurate stress levels. The finite element method of analysis is the proper tool for this task [22].

Many metal machine drives are subject to severe shock and impact loads. These loads are a major cause of severe damage and failure of drive train motors, gears, bearings and couplings. Also, they often cause rapid deterioration of support structures, loosening and breakage of foundation bolts and of the foundations themselves. By proper drive train design, modifying shafting and coupling torsional stiffnesses to tune the system, the effects of these loads can be dramatically reduced with a comparable reduction of maintenance and downtime [12]. It can be show [6] that the magnitude of the dynamic torque in the shaft system is related to the proximity of the system torsional natural frequencies.

In some of the more recent mills, however, difficulties have been experienced which could not reasonably be explained by reviewing the loads which were thought to exist during rolling. It was apparent that the stresses which exist in these drive components are associated with the phenomenon of torsional vibration. In the more recent installations, each mill stand is given the benefit of vibration analysis to determine the torque amplification factors of torsional vibration experienced during the normal rolling schedule and to calculate it and to reduce these factors to a specified upper limit [4].

Application of rolling load to the drive shaft has always been a matter of speculation. The torque to which a drive train are subjected with linear reaction should be well within fatigue limits of the components of the drive. The torques developed through nonlinear reactions are larger and can exceed the fatigue limits of the drive motor components. The practical way of analyzing the design requirements for these large peaks is through statistical study over a large number of ingots on several different mills to determine how often these high torques, which exceed the fatigue limit in the machine, occur [23].

In this work , it is aimed to construct a reliable mathematical model to study the motion and analysis of dynamic moments of torsional drive of machinery . The methods of analysis are the derived Lagrange's equations of the second order , and the mathematical modal analysis method in matrix form which are applied to obtain finally the results of this work .

2. FORMULATION OF THE PROBLEM

Consider a linear system of metal industry machine . the discs are made of rigid material and subjected to spring bindings with dampers and to external moments as shown in Figure (1).

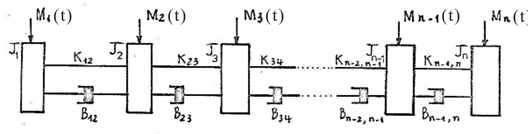

Figure (1) : Forced Damped Torsional Vibration Model.

The equations of motion for such system can be written in matrix form as [3],

$$\mathbf{J}\ddot{\varphi} + \mathbf{B}\dot{\varphi} + \mathbf{K}\varphi = \mathbf{M}(t) \qquad (1)$$

where:

φ - Column vector of angular displacement (rad.).

$\dot{\varphi}$ - Column vector of angular velocity (rad./s).

$\ddot{\varphi}$ - Column vector of angular acceleration (rad./s^2)

M (t) - Column vector of excited variable moments (N.m).

J -The square matrix of mass moments of inertia (kg.m^2) or (N.m)/(rad./ s^2) .

B - The square matrix of constants, characterizing linear dampings (N.m) / (rad./s).

K- The square matrix of torsional stiffness of springed connections (N.m / rad).

n - number of rigid discs.

The order of the matrices and vectors is equal to the number of degrees of freedom of the vibration mechanical system. Accurate result of solution of the mathematical model (eq.1) depends on many factors. From them that is produced the forced damped torsional model. The substance of this problem is to find out the necessary of degrees of freedom and the corresponding parameters of the model (J_i , B_{ij} , K_{ij}). For discuss of majority of this problem employ the method of reduction [3], [14] and [17].

Until the present time there are a row of authors solved the mathematical model (eq.1) by the modal analysis method for different metal industry machines [16], [17] and [24]. However, existence of the system for which assumed condition of modal analysis method, not only bring near the realization but it is necessary to select different process, example, solution of response by numerical mathematical methods [17] .For the best solution of the mathematical model (eq. 1) was often assumed that the moments of all motors have trapezoidal form (Figure 2).

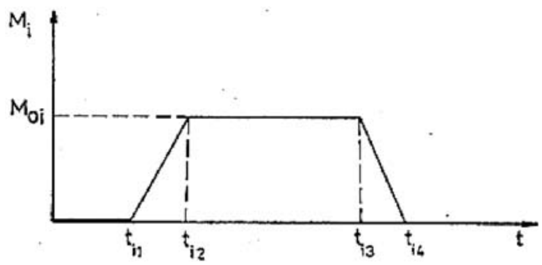

Figure (2): Moments of Electric Motors.

This assumption has only determined the authorization for the drives with disconnect regular loop of the velocities and with the motors which have small value of induction of armature. It is possible to know that the question of influence of the electrical phenomenon of the motors on the stress drived process are at this instant small workout for twin motors models [15].

Typical nonlinear mechanical models is the mechanical hysteresis (backlash) which expressed in some couplings and geared systems. By the way of backlash arise out the conditions for give rise to impulse moment which produces impulse of force. For the solution of dynamics of system with backlash considering concentration of all backlashes at these positions [14] and [8]. Solution of these systems (eq. 1) by conventional methods for linear systems are impossible. It is recommending to use harmonical linearization method or modeled of solution on the computer.

On the basis of study and analyses of the presented state of the calculation methods of the torsional vibrations of driving system of technological machine, the present work shows analysis of twin motor model as well as three degrees of freedom, and this with respect to the statical characteristic and dynamical characteristic of the driving electro-motors and on the character of the load on the metal industry machines.

The main task of this work is consideration of the influence of the electro-motor parameters on the torsional vibration of the drive twin motor model and from this also partial clear up the interaction between the electrical parameters of the electrical motors and the stress of the mechanical components.

In this work is a lot of use of linear theories neglecting the factors causing non-linearization the systems, also there are an analysis of the mechanical components of the metal industry machines from the point of technological view which is the influence of the dynamic torque amplification factors.

3. SOLUTION OF THE SYSTEM

In solution of the system (eq.1) by means of modal analysis, we may express the angular displacements φ(t) by means of new function ϕ(t) which is as function of time t. This is in matrix form will be:

$$\varphi(t) = \mathbf{V}\phi(t) \tag{2}$$

where,

V is modal matrix for the system equation (1) and

ϕ is called normal coordinate matrix of the system .

By substituting Equation (2) in Equation (1) and by premultipling the result equation on both sides by $\mathbf{V^T}$, Equation (1) reduced to the form:

$$\mathbf{V^TJV}\ddot{\phi} + \mathbf{V^TBV}\dot{\phi} + \mathbf{V^TKV}\phi = \mathbf{V^TM}(t) \tag{3}$$

or simply reduced to the form:

$$\mathbf{J_m}\ddot{\phi} + \mathbf{B_m}\dot{\phi} + \mathbf{K_m}\phi = \mathbf{M_m}(t) \tag{4}$$

where, the subscript m is indicated to the meaning of main

J_m= V^TJV = the main diagonal mass moments of inertia matrix and is called the modal mass matrix.

B_m= V^TBV = the main diagonal coefficient of dampings matrix and is called the modal damping matrix.

K_m = V^TKV= the main diagonal stiffnesses matrix and is called the modal stiffness matrix.

M_m(t) = **V^TM**(t) = The main load matrix (vector).

If we consider only the first disc from the system, then Equation (4) can be reduced to one single equation in the form :

$$J_1 \ddot{\phi}_1 + B_1 \dot{\phi} + K_1 \phi_1 = M_1(t)$$

$$\ddot{\phi}_1 + (B_1/J_1)\,\dot{\phi}_1 + (K_1/J_1)\,\phi_1 = M_1(t)/J_1$$

$$\ddot{\phi}_1 + 2\boxed{\delta_1}\,\omega_1 \dot{\phi}_1 + \omega_1^2\,\phi_1 = M_1(t)/J_1$$

where

J_1 = the first mass moment of iner

B_1 = the first coefficient of damping.

K_1 = the first torsional stiffness

$M_1(t)$ = the first load on the first disc.

$\boxed{\delta_1}$ = the first damping factor.

ω_1 = the natural torsional frequency of the first disc.

ω_1^2 = the squared natural torsional frequency of the first disc.

Now by premultipling Equation (4) by J_m^{-1} we obtain [24]:

$$\ddot{\phi} + 2.\delta.\lambda.\dot{\phi} + \lambda^2 \phi = J_m^{-1} M_m(t) \quad (5)$$

Where,

2. $\delta.\lambda$ = $J_m^{-1} B_m$, then

δ = [$\delta_1 \ldots \delta_k \ldots \delta_n$] = The diagonal square matrix of the damping factors.

λ = [$\omega_1 \ldots \omega_k \ldots \omega_n$] = The non-zero diagonal square matrix of the natural torsional frequencies of the system.

λ^2 = [$\omega_1^2 \ldots \omega_k^2 \ldots \omega_n^2$] = The non-zero diagonal square matrix of the squared natural torsional frequencies of the system.

Matrix Equation (5) introduces (n-1) independent linear differential equation system. The k^{th} row of the Equation (5) has the form:

$$\ddot{\phi}_k + 2\,\delta_k\,\omega_k\,\dot{\phi}_k + \omega_K^2\,\phi_k = J^{-1}_{mk} M_{mk}(t) \quad (6)$$

We can solve a number of the typical external moments in Equation (6) by use Duhamel integral and Duhamel tables [24]. For metal industry machine the external moments where often substituted by trapezoid form (Figure 2).

In this case, we have:

$$M_i(t) = \sum_{j=1}^{4} C_{ij}(t - t_{ij}).1(t - t_{ij}) \quad (7)$$

$$M_{mk}(t) = \sum_{j=1}^{4} C_{kj}(t - t_{ij}).\,1(t - t_{ij}) \quad (8)$$

where we established:

$$C_{Kj} = C_{ij}\ \varphi_{(oi)K} \quad (9)$$

$C_{i1} = \dfrac{M_{oi}}{t_{i2} - t_{i1}}$, $C_{i3} = \dfrac{M_{oi}}{t_{i3} - t_{i4}}$

$C_{i2} = - C_{i1}$, $C_{i4} = - C_{i3}$

and $1(t - t_{ij})$ is unit function , from which we write:

$(t - t_{ij}) < 0$	$1(t - t_{ij}) = 0$
$(t - t_{ij}) = 0$	$1(t - t_{ij}) = 1/2$
$(t - t_{ij}) > 0$	$1(t - t_{ij}) = 1$

Solution of Equation (6) by assuming Equation (8) possible leading by the use of Laplace transformation we express that in matrix form as [3]:

$$\phi = \exp(-\delta_k\, \omega_k\, t).\cos[\omega_k(1-\delta_k^2)^{1/2}.t].\,\phi(o) + \exp(-\delta_k\omega_k\, t).\,(1-\delta^2)^{1/2}.\,\lambda^{-1}.\,\mathrm{Sin}[\omega_k(1-\delta_k^2)^{1/2}.\,t]\,[\dot{\phi}(0)+\delta.\lambda.\phi(0)] + J_m^{-1}.\lambda^{-2}.\,F_k\delta \quad (10)$$

where we know:

$\exp(-\delta_k\,\omega_k\, t) = [\, e^{-\delta_1.\omega_1.t} \dots e^{-\delta_k.\omega_k.t} \dots e^{-\delta_n.\omega_n.t}\,]$

$\mathrm{Cos}\,[\omega_k(1-\delta_k^2)^{1/2}t\,] = [\,\mathrm{Cos}[\omega_1(1-\delta_1^2)^{1/2}t\,] \dots \cos[\omega_n(1-\delta_n^2)^{1/2}t\,]\,]$.

$\mathrm{Sin}\,[\omega_k(1-\delta_k^2)^{1/2}t\,] = [\,\sin[\omega_1(1-\delta_1^2)^{1/2}t\,] \dots \sin[\omega_n(1-\delta_n^2)^{1/2}t\,]\,]$

$F_k\delta = [\,F_1\delta \dots F_k\delta \dots F_n\delta\,]$ … column matrix of response functions, their elements are:

$$F_k\delta = \sum_{j=1}^{4} C_{kj}\,\{t - t_{ij} - T_k + e^{-Nk(t-tij)}\,[A_k\,\mathrm{Sin}\,\Omega_k(t-t_{ij}) + T_k\,\mathrm{Cos}\,\Omega_k(t-t_{ij})]\}.1(t-t_{ij} \quad (11)$$

and here: $N_k = \delta_k\,\omega_k$ $\quad\Omega_k = (\omega_k - N_k^2)^{1/2}$

$T_k = 2\delta_k\,\omega_k^{-1}$ $\quad A_k = (2\delta_k^2 - 1)\,\Omega_k^{-1}$

$\phi(0)$ and $\dot{\phi}(0)$ are constant calculated vectors.

For zero conditions, $\varphi(0) = 0$, $\dot{\varphi}(0) = 0$

Vector of displacements $\varphi(t)$ of system we obtain transformation (eq. 2) in the form:

$$\varphi(t) = V.\,J_m^{-1}.\,\lambda^{-2}.\,F_k\delta \quad (12)$$

In this work we used the numerical computation method. In the case of consideration of the influence of the electrical phenomena of the electric motors we cannot reduce the matrix Equation (1) into independent (individual) linear differential system. It is possible that we solve the system (eq.1) on the basis of numerical computation method, that every ordinary linear differential equation with n^{th} order we can add to it a certain ordinary differential equation with one order, thus the solution of equations with n^{th} order is determined by the single valued solution relevant to the system.

In this work, we calculated the dynamic response of the twin motor model with respect to the transient response in the electric motor, parameters of this model are represented in Figure (3). We have used PROSA applying the Runge-Kutta method with 4^{th} orders. Thus the method of solution is explained as follows, so that from sizes with 20 elements used in work [24] and others the reduction method we draw up with twin motor model with three degrees of freedom (Figure 3).

Now we can reduce the mechanical system shown in Figure (1) which is represented in the mathematical model in Equation (1) to twin motor model with three degrees of freedom (Figure 3).

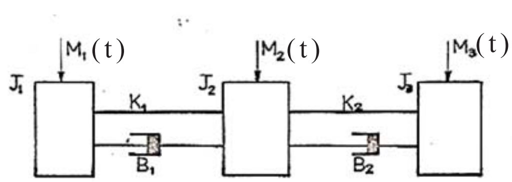

Figure (2): Moments of Electric Motors.

where, M_1, M_3 = moments of electro-motors .
M_2 = load moment.

The equations of motion have the form:

$$J_1\,\ddot{\varphi}_1 + B_1(\dot{\varphi}_1 - \dot{\varphi}_2) + K_1(\varphi_1 - \varphi_2) = M_1(t)$$

$$J_2\,\ddot{\varphi}_2 + B_1(\dot{\varphi}_2 - \dot{\varphi}_1) + B_2(\dot{\varphi}_2 - \dot{\varphi}_3) + K_1(\varphi_2 - \varphi_1) + K_2(\varphi_2 - \varphi_3) = M_2(t)$$

$$J_3\,\ddot{\varphi}_3 + B_2(\dot{\varphi}_3 - \dot{\varphi}_2) + K_2(\varphi_3 - \varphi_2) = M_3(t)$$

(13)

Solution of Equations (13) is practically for unit moment of load M_2 and for the period of

time 0.005 second to 0.5 second for this moment. We assumed that the initial conditions are:

$$\varphi_{10} = \varphi_{20} = \varphi_{30} = 0$$
$$\dot{\varphi}_{10} = \dot{\varphi}_{20} = \dot{\varphi}_{30} = \omega_{on} \qquad (14)$$

Where, ω_{on} is idly velocity of the electric motors.

From this calculation of conditions there are five alternative solutions:

1. Loading moment M_2 method only on the system.
2. Synchronous M_1, M_2, M_3 method on this system.
3. Without synchronous M_1, M_2, M_3, method on the system.
4. Response the system with influences of the hardness characteristics of electric motors.
5. Response of the system with influence of induction armature of the electric motors.

From 1 to 3 we assume that M_i (i = 1,2,3) have trapezoid form.

For solution of numerical computation method we reduce the system (eq.13) on the system of ordinary linear differential equation with one order. The algorithm for the solution was written in FORTRAN LANGUAGE.

For alternative 1,2,3 considering linear course of loading moments. For alternative 4 we take:

$$M_i = \bar{\bar{M}}_k - \bar{\bar{\beta}}\,\dot{\varphi}_i \qquad (15)$$

where, i = 1,3 and $\bar{\bar{M}}_k$ is the moment for a short time of electro-motors and $\bar{\bar{\beta}}$ is the hardness characteristics of electro-motors .

In the case with reason of induction armatures of the electric motors (for alternative 5) we have: $M_i = C\phi \cdot i_i$

$$\dot{i}_i + \frac{1}{\tau_a} i_i = \frac{U_a}{L_a} - \frac{C\phi}{L_a}\dot{\varphi}_i \qquad (16)$$

where,

i =1, 3, i_i is the electrical current, Ampere unit, U_a is the electrical voltage,

τ_a is the electromagnetic time constant,
L_a is the induction circuit of armature and
$C\phi$ is the construction parameter of electro motor.

4. RESULTS AND DISCUSSION

Results of calculations are given in a group of graphs starting from Figure (4) to Figure (13).

1. The timed course of the torsional moments M^k_{12}, M^k_{23} (index K indicates to K[th] alternative and K=1,2,3,4,5) is with effect of the damping vibration.
2. In the reducing zone (area) of the absolute values of M^k_{12}, M^k_{23} (K = 1,2,3,4,5) reach the maximum in the same point (t = 0.03 second). It means that for the mechanical elements of the connecting shafts with dangerous instant getting late than instant, in which of all the external moments reach to the maximum values. At the same time to show, that the electrical reduced phenomena of the driving motors do not have influence on this instant. For K = 3 this instant is t = 0.1 second. In this instant M_1, M_3 reach at the same time to the maximum value. Here the dynamical response of the system examine the velocity.
   Values of the dynamical factors β^k_{12}, β^k_{23} (K = 1, 2, 3, 4, 5) show the particular difference. From all the alternatives' cases K = 3 is very bad. In this case not only β^3_{12}, β^3_{23} have the biggest values, but also the most unfavorable forms of the reduced zones (areas). In the cases K = 1 and K = 2 we have $\beta^1_{12} = \beta^2_{12}$, $\beta^1_{23} = \beta^2_{23}$. This lead to the conclusion that in the case K = 2 the dynamically of the system is as in the case, when only the way of the load of the moment M_2 on the system.
3. Influence of factors of the hardness of characteristics $\bar{\bar{\beta}}$ and electromagnetic actions (respecting study of induction armatures motors) is expressed as the minimizing values of the dynamical

factors. Here β^{5}_{12}, β^{5}_{23} are less than β^{4}_{12}, β^{4}_{23}.

In this case of K = 4, the influence of $\overline{\beta}$ is clearly expressed the shortness time of the reduced zone (area) of the external moments M^4_{12}, M^4_{23}. There is bigger $\overline{\beta}$, and short time. From these cases, the investigation of the dynamical responses with respect to $\overline{\beta}$ gives us not quite exact course reduced phenomenon than what we have from experimental measurement. At the same time we state that the result of K = 5 the biggest approach to the value as well as to the experimental result of the course of external moments of the system. Generally it is possible to state that the capability of the damping system will increase the consideration of the influence of reduced of phenomena of the motors.

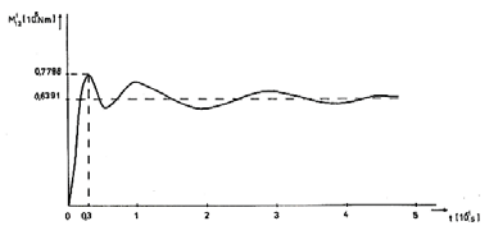

Figure (4): Course of Torsional Moment $M^1{}_{12}$.

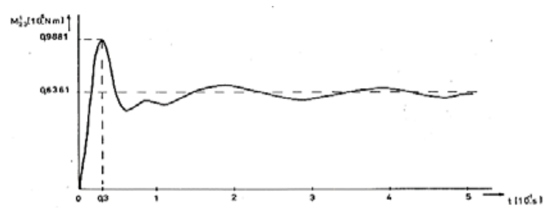

Figure (5): Course of Torsional Moment $M^1{}_{23}$.

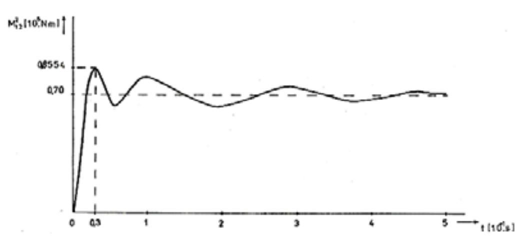

Figure (6): Course of Torsional Moment $M^2{}_{12}$.

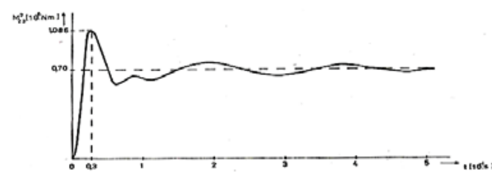

Figure (7): Course of Torsional Moment $M^2{}_{23}$.

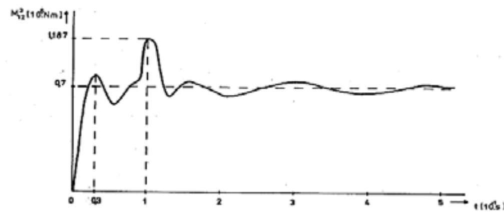

Figure (8): Course of Torsional Moment $M^3{}_{12}$.

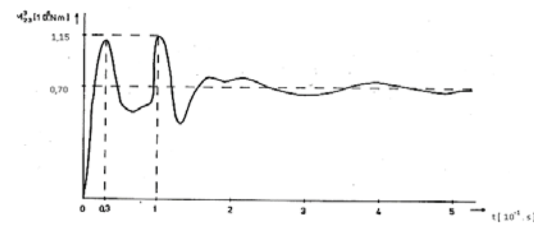

Figure (9): Course of Torsional Moment $M^3{}_{23}$.

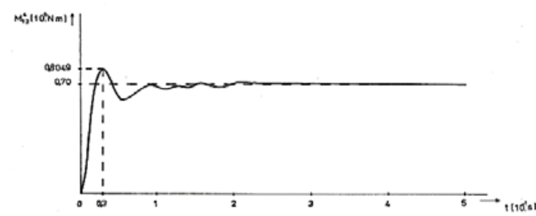

Figure (10): Course of Torsional Moment $M^4{}_{12}$.

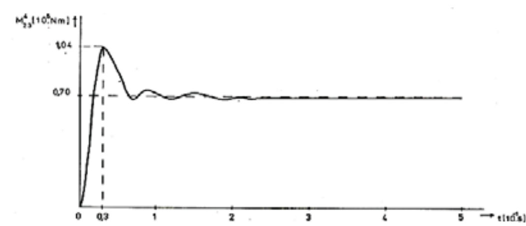

Figure (11): Course of Torsional Moment $M^4{}_{23}$.

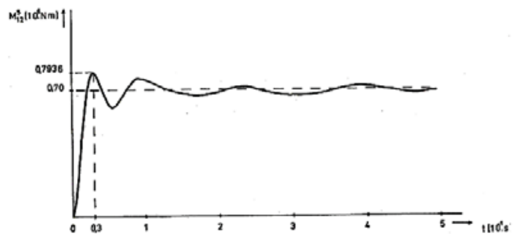

Figure (12): Course of Torsional Moment $M^5{}_{12}$.

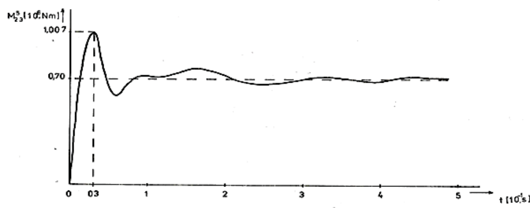

Figure (13): Course of Torsional Moment $M^5{}_{23}$.

5. CONCLUSION

Present requirements on increase of the operational reliability of metal industry machine leads the solution of the problematics of analyses of the dynamical consequences of the drive of this arrangement have big significance.

The expanded work which aims to endeavours the analysis of the dynamical response at unregulated loading of the twin motor model with three degrees of freedom (respecting metal industry machine, Figure 3).

The obtained results of calculations facilitate partially to evaluate of the influence parameters of direct current of electro motors with separate excitation on magnitude and course of torsional moments in several cases of external moments. They were proving the possibility of damping of torsional vibrations of this model by means of parameters of electric motors. At the same time it follows that there exist possibility to reduce the dynamical stress of drived system of electrical parameters. In this direction it is necessary to give investigation, first provided about regulated model, may be more clarify the mutual relationship between the reducing (transient) of electrical and mechanical phenomena.

Comparison with earlier results and from expanded work it is still unmistaken, that is neglecting the influence of factors of hardness of electric motors and also the replacement of dynamic characteristics (statics) may be lead to not corrected results. We can therefore, state that neglecting the influence of characteristics of motor may not be generally admitted.

At the same time it may be stated that used methods (modal analysis method resp. numerical mathematical method) are generally utilizabled to investigate dynamics of drive of different types of arrangements.

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