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COMPUTATIONAL DYNAMICS OF HELICAL FLEXIBLE COUPLING WITH TRANSITORY CONTINUOUS REGIME

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Abstract: *Dynamic behaviour of flexible couplings within the vibratory equipments frames the global area of this study. This paper briefly presents some critical results regarding the helical-type coupling dynamics taking into account a few theoretical assessments in terms of linear and nonlinear approaches of driving chain elements. Simulated behaviors denote that many of the reported observation regarding the serviceable dynamic distortions are correctly, and impose a set of additional instrumental tests in order to obtain a good accuracy for mathematical models. A both simplified and expanded computational model has used for numerical computations. The main hypothesis of this analysis supposes the continuous changing of dynamic characteristics, and the additional dynamic loadings due to internal elements of coupling device.*
Keywords: *Computational dynamics, Helical flexible coupling, Transitory state, Vibration, Functional optimization*

1. INTRODUCTION

Theoretical approaches, numerical simulations and instrumental investigations of systems with vibratory equipments driven by rotary electric motors had revealed that helical flexible coupling although provides lot of advantages in regular utilization, but can also provide a major source for transitory dynamics into the driving system. Various distorting signals acts such an inputs and perturb the technical system thus that it is quite difficult to identify, estimate and evaluate them just based on instrumental tests [2-6]. Hereby, appears the necessity of a suitable mathematical model that had to have implemented into a computational environment and used for analysis and estimations of dynamic behaviour of elastic coupling [3,5].
The second consideration sustaining this study takes into consideration the large area of mechanical transmissions provided with flexible coupling devices having various structural and functional characteristics. Photo in Fig. 1 depicts two types of helix-type flexible couplings, supplied by the Helical Products Company, Inc., USA, and this shows but one of the available type of flexible coupling devices all over the world.

Figure 1: Helix-type flexible coupling devices
[*from Web-catalog of Helical Products Company, Inc., USA*]

The computational analysis briefly presented in this paper has started with the basic approaches proposed by the authors in the previous study [3] and has based on the next main hypothesis as follows:
- The dynamic nonlinearity of torsional spring zone within the helical flexible coupling, assuming that enlarging or reducing of coil diameter can induces modifications of the dynamic characteristics because of the continuous changing of dynamic masses distribution during a complete rotation cycle [2].
- The variances of rotational speeds at the coupling input and output respectively, induce certain abnormal dynamic evolution in terms of functional deformation, globally or locally arising, taking into account that positive or negative applied bending moment means changing of the diameter onwards or backwards with respect in direction of coil winding [4].

- Inside the torsional spring area, can appears resonances, locally into a certain coil curl or a group of, or globally into the spring on a whole. Effective conditions leading to a resonant phenomenon in coupling coil cannot be strictly evaluated because of continuous changing both of the internal system parameters (dynamic mass distribution and stiffness), and of the rotational speed (actually the variances of the velocity through the coupling spring, between input and output clamping bushes).

2. ANALYTICAL APPROACHES OF FLEXIBLE COUPLING DEVICES

A quite simple model for a driving system such as *motor ↔ coupling device ↔ load* has depicted in Fig. 2. In respect with the initial hypothesis according with the motor will have simulated such a real or an ideal mechanical power supply element, results the system of two dynamic equations or a single one respectively (the model have two degree of freedom corresponding the real case, and a single degree of freedom corresponding the ideal case).

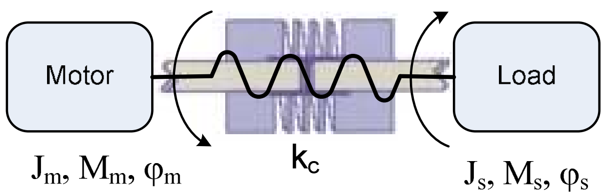

Figure 2: Simple lumped mass model for flexible coupling driving system

Obviously, this simple model includes a single way to analyze the flexible torsional coupling dynamics that is the stiffness expressions for the round or for the rectangular wire torsional springs respectively, as follows

$$k = \frac{\pi}{180}\frac{E\,d^4}{64\,n\,D}\;;\; k = \frac{1}{180}\frac{E\,b\,t^3}{12\,n\,D}\;;\;[\text{deg}]. \tag{1}$$

In Eqn. (1) the parameter E denotes the modulus of elasticity, d is the spring wire diameter, D denotes mean coil diameter, n is the number of the active coils, b is the wire width, and t is the wire thickness. First fractional terms in both expressions in Eqn. (1) help to obtain the stiffness value in deg units.
An extended approach of the model in Fig. 2 must be providing by nonlinear laws for torsional rigidity. This kind of model even it will be able to accomplish a realistic simulation, also requires difficult evaluations for many parameters and assumes higher order computations capabilities [1,6].

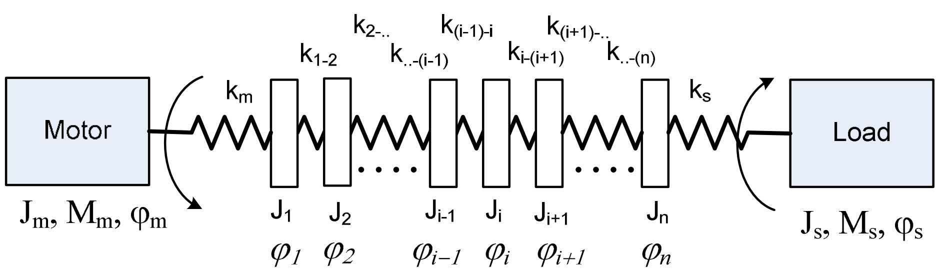

Figure 3: Complete lumped mass model for flexible coupling driving system [3]

Next step on analytical approach and numerical implementation supposes a model with lumped mass schematization for the coupling device thus that it will be able to dignify the way to behave for each elementary part of flexible coupling [2,3]. In Fig. 3 was depicted a complex lumped mass model for a helical flexible coupling driving scheme.
The second model (see Fig. 3) can provides ($n+1$) or ($n+2$) degrees of freedom (DoF) according with the initial hypothesis for the motor device simulation complexity (with ideal or real power supply characteristics). The basic nDoF corresponds to the number of the elements assumed for helical device sampling, and the load provides the additional DoF. A matrix formulation of dynamic equations is [3]

$$\mathbf{J}\,\ddot{\varphi} + \mathbf{K}\,\varphi = \mathbf{F} \tag{1}$$

where **J** denote the matrix of inertia, **K** is the matrix of rigidity, **F** is the external forces, and φ denotes the angular displacement.

The matrix of inertia **J** is a diagonal array containing the moment of inertia for each element of the system, including additional inertia due to the dynamic effects in any coil of coupling spring.
External forces column vector denoted by **F** contains the external resistant or supplying moments of forces for the entire systems. In case of hypothesis of the resistant moment of forces developed by the instant dynamic eccentricity of a single or a group of coils, those values have to written also into the vector **F**.
Damping matrix is missing from the Eqn. (2) because even if it assumes Rayleigh-type proportional dissipative component approximation, or it supposes loss factor-type damping, both hypotheses does not provide any additional information for system response in terms of the output signal spectral composition.
Nonlinear characteristics for stiffness can has assigned for any of the linkages but it had to taken into account the remarks at beginning of this paragraph regarding the nonlinearity of torsional rigidity and the additional aspects that involved.
Matrix of rigidity, noted by **K** in Eqn.(1), is a real symmetrical array contains the stiffness value of every linkage inside the model and can be defined as follows

$$\mathbf{K} = \left\| K_{i,j} \right\|_{\substack{i=1\dots n \\ j=1\dots n}} = \begin{vmatrix} k_{i-1} + k_i & \textit{in case of} & i = j \\ -k_i & \textit{in case of} & \begin{cases} j = i+1 \\ i = j+1 \end{cases} \\ k_0 = 0 & & \\ k_{n+1} = 0 & & \end{vmatrix} . \tag{2}$$

Results evaluation for this model assumes very large systems with big number of parameters to accomplish, to analyze and to present. For usual cases of analysis it can be simulated the flexible helical coupling like two separate linear linkage elements, with half stiffness, mounted between the motor/load components and a central virtual rotational body included full inertia characteristics of coupling.

3. COMPUTATIONAL APPROACH AND DISCUSSIONS

Diagrams depicted in Fig. 4 and Fig. 5 shows the evolution of the load and coupling angular velocity and the spectral composition of each signal. Actually, it has drawn the deviation velocity between the variable outputs and the input fixed value of angular speed at motor shaft.
Graphs in Fig. 4 denote a few seconds timed evolution of the system supposing a regular dynamic behaviour. This means a very rigid coupling with relative low mass and no additional eccentric effects both at load and inside the coupling spring. Missing of the additional inertial effect at the load can have assigned to the situation while the vibratory equipment that works into the horizontal plan.

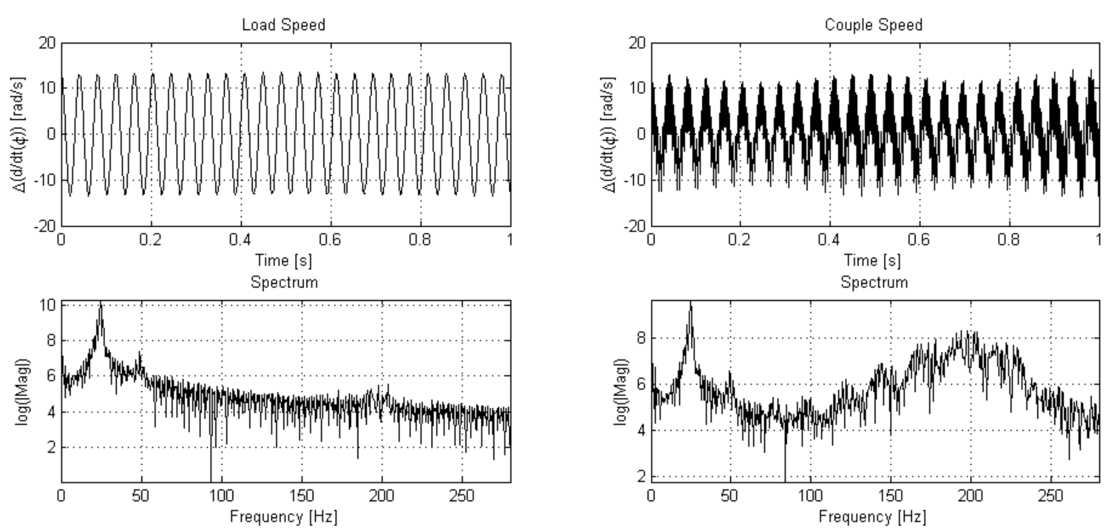

Figure 4: Steady state evolutions of driving system parameter

The graphs depicted in Fig 5 presents the same parameters as in Fig. 4 but for a strong transitory case meaning considerable mass and additional inertial effects in the entire coupling element, and the presence of the load eccentricity effect. Time length for the second test has enlarged to dignify that the coupling evolution is not at

the resonance and its velocity amplitude even if acquiring a periodically great values its variance has finite limits.
Comparative analysis between the two sets of diagrams has to taken into account different excitation for each case. Although, it easily results a "silenced" evolution in the first case and in the same time a relative "noisy" behavior for the second one. Additional dynamic effects induce a frequency modulation of general movement with very strong impact into the coupling coils. Most significant peaks were marked on the spectral composition diagrams. Each pairs of spectrum graphs shows that the peaks has preserved for the same case (acquire the same frequency values). However, it also preserved the global trend, which reveals that the loads acquire maximum magnitudes for low frequencies and, in the same time, the flexible coupling slides to the higher values of frequency for relevant magnitudes.

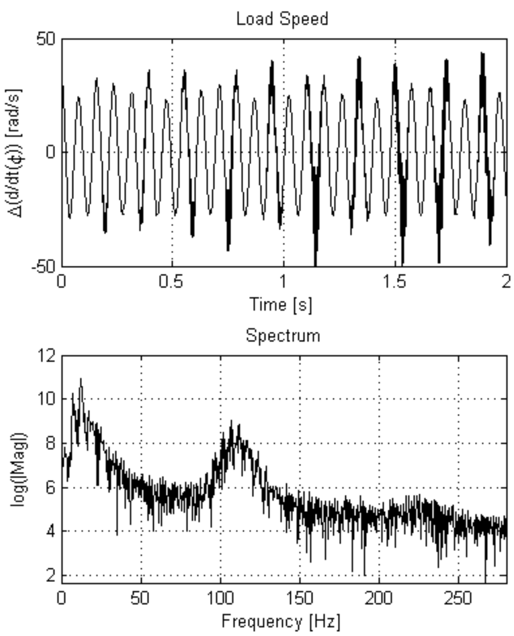

Figure 5: Transitory state evolutions of driving system parameter

4. CONCLUDING REMARKS

Basic approaches presented in the paper both enable and supply the future developments of the helix-type flexible coupling behaviour and its dynamic influences on the global evolution of the entire actuating systems.
The briefly presented results have shown the differences between spectral compositions of the load movement for two extreme cases and have revealed the opportunity of such analysis.
Comparative and correlative analysis of the whole set of numerical results but taking into account only the first approximation for this computational research has been dignified strong necessity of multi-body approaching for the active part of coupling device. In addition, has revealed the opportunity of using the real characteristics for both the input (the source of energy) and the output (the load) of the system in order to have an accurate simulation and to dignify the feedback character of the source/load elements whatever the exploitation regime.
The experimental area of this research will have completed with a set of instrumental tests performed on a special laboratory stand with the purpose of evaluation for different cases of dynamic regimes and their influences on the load evolution.

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