RESULTS OF CALCULATIONS AND MEASUREMENTS OF TORSIONAL VIBRATIONS*

By

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The invitation of the Department of Technical Sciences of the Hungarian Academy of Sciences to function as co-reporter with Dr. Ádám BOSZNAY, University Professor, I received with genuine pleasure.

BOSZNAY in his paper reported on systems with finite freedom degrees: the movement of many real systems can be readily described with systems of differential equations relating to a discrete model.

To this statement, whose validity is incontestable, I wish to add that, if we accept the standpoint that basic and applied researches (the latter having great importance in industry) cannot, and must not, be separated but much rather interrelated, the present and future industrial structure imperatively calls for intensive research into the theoretical and practical problems of vibrating systems having finite freedom degrees. So much about the topicality of the subject.

And now it will briefly be spoken about the results we have obtained in our examinations into torsional vibration systems.

As is known, the examination of torsional vibration systems consists of the following principal parts:

1. The evolution of the model of a given vibration system. This includes:
   a) the reduction of mass and length;
   b) the analysis of excitations;
   c) the analysis of dampings.
2. The determination of the points of resonance and the pattern of vibrations, including:
   a) the determination of the natural frequencies, resp. the pattern of natural vibrations;
   b) the detemination of harmful harmonics in the light of the range of working velocities;
   c) the calculation of the so-called resonance stresses at the resonance levels associated with harmful harmonics.

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3. Torsiography by means of supervision, which yields information on:

a) the accuracy of mass and length reductions;

b) the need (if any) to calculate non-resonance stationary vibrations;

c) the need (if any) of modifications in the vibrating system, viz. the use of an optimum system, or else, of the application of some device of limiting vibration.

4. The calculation of the requisite modifications, viz.:

a) variations in the spring or in the mass;

b) the dimensioning of a vibration detuner or a vibration damper.

The sequence may naturally be somewhat varied, depending on whether the examinations serve the designing of a new equipment or the improvement of an existing one.

Each part of the calculations is greatly work consuming and it may be due to this fact that industrial enterprises, accustomed to quick and concise calculations, do not resort to such calculation methods unless forced to do so by special circumstances. On this consideration in recent years we endeavoured to use electronic computer for the partial calculations.

We have evolved universal programmes which ultimately yield all partial and final results in such a way that the result of the first calculation is carried on to the tape of the second, and so on. In the sequence of examinations, these programmes are as follows:

1. On the basis of the technical drafts of some equipment or vibration system, the reduction of the mass and length as well as the evolution of the vibration model are carried out according to computer programme. This programme at the same time computes weights and sectional moduli. We intend to train experts of the interested factories in the performance of mass and length reductions along the given programme.

2. On the basis of the indicator chart of some internal combustion engine, considering the dimensions of the crank mechanism, we carry out the analysis of excitation by computer programme. It yields the tangential and radial forces in ordinates of optional spacing — for instance 20, 10 or 5° apart, etc.

3. The natural frequencies, natural vibration patterns, amplitude differences and the sum of the squares of amplitudes have, for quite some time, been calculated according to computer programme, based on the Holzer—Tolle system and built up in such a way that the residual torque is printed out. In most cases it gives zero and indicates errors in the approximation.

4. To calculate non-resonance stationary forced vibrations, a programme is being drawn up in consideration of dampings, excitations and non-linearities. Over and above this programme, we are currently working on a similarly complex synthetizing programme.

At this point the importance of BOSZNAY's research work must be underlined, since it enables the synthetis of systems with predetermined natural frequencies [1]. Its further development is desirable, though, since in the synthesis of torsional vibration systems the determined point of node or the point of maximum stress are stringent criteria, which are equivalent in importance to the predetermined frequencies.

5. Also the harmonic analysis of the measurement results has become necessary. For this purpose a programme has been devised which computes all possible (n — 1) Fourier coefficients of periodic curves determined by optional 2n ordinates.

6. We have elaborated several suitable programmes to calculate the required modifications.

We have evolved, first of all, a so-called optimizing programme to calculate the required, respectively, the optimally modified, system which even without the use of a separate damper, produces vibrations less than the permissible limit.

Theoretical examinations verified by measurements, brought about very good results along the programmes optimizing with the said two parameters, for a group of the multi-mass torsional vibration systems. These vibration systems are characterized (or can be retraced) by a single-connected chain in which the dominant excitation is given at a certain number of masses (*e.g.* 4, 6, 8), and dominant damping is concentrated at some points (*e. g.* 1, 2, 3). Such, for instance, is the vibration system of a 1500-ton seafaring vessel which we had the opportunity of examining.

On this basis — and this has been verified also by plant tests — we may assume that in the newer vibration systems obtained by varying the mass and the spring, the magnitude of excitation resp. damping will show very slight variations. These, therefore, should not be regarded as optimization parameters either. One of the optimization parameters has been evolved from the correlation between the points of resonance caused by the dominant excitation harmonics and the range of working velocities, while the other was formed from the so-called dimensionless stress calculable from the identity of the natural vibration pattern and the resonance vibration pattern in which we also included the permissible stresses.

We have performed numerous examinations with this programme. They yielded highly interesting results, on which we will report in another paper [4].

Secondly: We have computer programmes for the dimensioning of two damper devices: 1) for a linear and 2) for a non-linear pendulum vibration detuner.

In our examinations — theoretical as well as practical — we used the non-linear detuner programme predominantly.

We have elaborated the method for the dimensioning of the non-linear vibration detuner [2]. On the basis of calculations a mechanism has been designed and put in to operation since, which verified the correctness of the method.

In our work the Elliot 803 B-type digital computer of the Board of Iron Metallurgy of the Ministry of Metallurgy and Engineering was used.

To measure parameters which cannot be accurately calculated or which are strongly non-linear, in our research into the migration of nodes and for various other purposes, we have designed the SB—1 equipment. In its design principles it is based on a similar equipment evolved by Professor L. I. STEINWOLF at the Department of Dynamics and Stability of Machines at the Charkow Polytechnical University.

Torsional vibration measurements have been carried out in recent years also under plant conditions, among others on railway diesel engines and marine diesels, in test runs. In addition to the classic type of the still widespread Geiger mechanical torsiograph, we made use of our own ET—1 torsiograph with tensometric amplifier. For the dynamic calibration of the ballistic measuring heads, a special device was assembled.

To increase measurement accuracy we have elaborated a new and up-to-date digital instrument in cooperation with Mr. Péter THEISZ. The calibration and testing of the instrument are in progress.

The instrument will permit measurements to be performed not only on the free shaft ends but also at intermediate cross sections. This facility, higher accuracy and the fact that the instrument yields information (consisting of a great number of data — 720 per period —) printed on punched tape, which can readily be processed in electronic computer, hold out good promise for its use. We have elaborated a suitable programme also for mechanical data processing.

As a conclusion, a few words on these promises.

Fast computer programmes, measurements which can be accurately and rapidly assessed, and systematized basic data and measurement results (particularly the latter) have induced us to urge the introduction of TMD (the Hungarian initials for Scientific and Technical Diagnostics) in theory as well as in practice.

Experts are familiar with the fact that on machines, kept in good working order and intimately known by their operators, from noises the imminent failure of some component can be predicted, excessive wear and defects established. Accurate vibration measurements may tell even more about the actual state of a machinery.

Diagnoses relying on these measurements are particularly important in the final checking of power machines, vehicles, various other machines and equipment. This subject is treated in a more detailed manner in a separate paper [3].

Until such a time as this method can be introduced extensively, a great deal of theoretical research, experiments and running tests will be necessary.

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References

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