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A new damper to solve galloping on bundled lines. Theoretical background, laboratory and field results.

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Abstract - A new kind of anti-galloping device for bundled lines is presented. The approach is complete, including basic mechanisms, laboratory results, valuation of the efficiency on a whole line and field results.

I. INTRODUCTION

The use of bundle conductors arrangements for the transmission of electrical energy is until now and by far, the best technical and economical solution. It remains, however, that this kind of large structure because of its highly flexible form is very sensitive to wind exposition. The observations allow to identify three main wind-induced vibrations, namely : aeolian vibration that is caused by an alternating wind force which arises from a difference of pressure associated with a regular formation of vortices behind a conductor, wake-induced oscillation that is produced by forces from the shielding effect of windward subconductors on their leeward counterparts and finally the galloping vibration which is always caused by moderately strong, steady crosswind acting upon an asymmetrically iced conductor surface.

The main characteristics of galloping are low frequency (from 0.15 Hz to 1 Hz for a typical extra high voltage construction) and large amplitude (from 0.1 to 1.0 times the sag of the span) with a single or a few loops of standing waves per span.

The effects of galloping on a line are dependent on the severity and duration of the event and on the type of line construction. Typical problems are : flashovers causing circuit breaker operation and arcing damage to conductors, including occasional conductor failure, infringement of clearance to ground, loosening and ejection of tower bolts, fatigue of tower steelworks during sustained events, conductor fatigue at tension strings, jumper fatigue, damage to spacers and stockbridge dampers, possibility of string failure, damage to conductor strands at suspension clamps and spacers clamps from fault current equalisation within the bundle. All these problems require high annual average cost for repairs, for the building of lines to have large clearance, as well as for the installation of control devices.

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Since the behaviour of bundled lines in torsion is quite different from the one of single conductors, the instability for what concerns galloping of bundles cannot be explained by the classical aerodynamic instability criterion of Den Hartog [3].

Based on this assumption, this paper presents a new damper, specially designed to solve galloping on bundled lines.

A systematic approach is presented starting with the review of the main mechanisms which govern galloping of bundle conductors and the already existing anti-galloping devices. A case study is also introduced for the whole line system in order to enlighten the presented mechanisms.

Then a detailed description of the damper is given together with its dynamical requirements and manufacturing story.

Afterwards the different laboratory results are presented. They concern the high-voltage behaviour of the device tested in the installation of Laborelec in Belgium and the dynamical behaviour evaluated in the laboratory of Professor Hagedorn at the Technische Hochschule Darmstadt in Germany.

Next the results acquired in Darmstadt are transposed on a real line to assess the efficiency of the new device.

Finally promising field results obtained on a line in Belgium are discussed.

II. STATE-OF-THE-ART

A. Mechanisms

In opposition to single conductors, bundled lines possess a torsional stiffness [8] that prevents significant rotation from the icy sheath during its formation. This situation leads to a very eccentric sheath for which the Den-Hartog condition in general does not hold. Indeed, measurements carried out on real ice profiles [1, 5, 6] suggest that the Den-Hartog condition holds only for very slightly eccentric sheaths with lift characteristics opposed to those of more markedly eccentric profiles.

However, the natural vertical and torsional oscillation frequencies of spacered bundles are in the same narrow range. This is not a chance occurrence due to line dimensions but a structural property of bundles. Under these conditions, despite torsional stiffness, a resonance phenomenon may cause vertical oscillation to excite torsion of sufficient amplitude and suitable phase to de-stabilise the system. The physical mechanism is in many ways similar to the one which caused the collapse of the Tacoma bridge and to the "fluttering" of airplane wings, well-known to experts in aerodynamics [1, 2, 5]. This phenomenon is thus called "flutter galloping".

There is no way to explain in one page the complex theory of flutter galloping. This has been done in the literature by numerous authors [4, 7]. The mechanism of galloping is very complex and the impact of torsional damping has not been well understood in the major part of the literature: some authors did not see the fundamental influence of torsional damping because there is no possibility to damp the galloping energy which is mainly in the vertical movement. But in fact torsional damping plays a key role on the angle of attack (position of ice eccentricity in relation to relative wind speed) hence preventing energy to be inputted on the line so that either instability is avoided or - for very high wind and big eccentricity - amplitude is sensitively reduced. So the combination of **detuning** (shifting between vertical and torsional frequencies), which avoids resonances, and **torsional damping**, which limits transfer of energy, seems to be the solution of the flutter galloping problem. This is possible on overhead lines because the amount of torsional energy to be dissipated is rather low (some tenths of joules per span).

Two important anti-galloping devices for bundle conductors exist until now on the market : the detuning pendulums [6] (Canada) and the GCD (galloping control device) [9] (Japan). The detuning pendulum only increases the torsional frequency and adds no damping to the phenomenon. This is an important limitation which requires a perfect design of the pendulums. Generally this is not possible because ice conditions are not known at the design stage. The GCD device increases the moment of inertia of a bundle line, without any effect on torsional stiffness and with negligible damping. This is another type of detuning by decreasing torsional frequencies.

B. A case study

The aim of this section is to introduce a case study in order to illustrate the notions mentioned before and to assess the efficiency of the new damper described below.

The galloping phenomenon can be studied by standing waves decomposition limited to maximum three loops for each movement (vertical, transversal and torsional) and for each span. This is the approach used by J.Wang in his very complete Multi-Span 3-DOF model [10].

For the purpose of this paper we will only use a very simplified form of J.Wang's results by neglecting the couplings between different spans and modes and by considering only the first vertical and the first torsional mode of a horizontal suspended twin bundle conductors. In this case it can be showed that the free oscillations frequencies of the line around his equilibrium position are governed by

$$\begin{cases} f_v = \dfrac{1}{2L}\sqrt{\dfrac{T_0}{m}} \\ f_\theta = \dfrac{1}{2L}\sqrt{\dfrac{GJ}{I}} \end{cases} \quad (1)$$

$$\text{with} \quad GJ = \tau + r^2 T_0$$

where r is the radius of the bundle, T_0 is the sagging tension, τ is the intrinsic torsional stiffness of the bundle, GJ is the total torsional stiffness of the bundle, m and I are respectively the total mass and the moment of inertia of the iced bundle's cross-section per unit length and L is the length of the span. If we try data from a 420 kV horizontal twin bundle transmission line that was subjected to galloping in Norway, namely : $r = 0.45$ m, $T_0 = 101604$ N, $\tau = 1060$ Nm^2/rad, $I = 1.166$ kgm, and $m = 5.76$ kg/m, we have for the ratio between the torsional and the vertical frequency:

$$\frac{f_\theta}{f_v} \cong 1.03 \quad (2)$$

that confirms the structural characteristic mentioned before.

III. A NEW CONCEPT, THE TORSIONAL DAMPER AND DETUNER

A. Description

The new damper, called Torsional Damper and Detuner (TDD) is a practical device which combines **a detuning effect** with a **high torsional damping**. This concept can be applied to every type of bundle configuration (every number of subconductor). Nevertheless the available results were obtained on a prototype designed for horizontal twin bundled lines.

A schematic representation is given on Fig. 1. The whole system is attached to the twin bundle with four clamps and can move in rotation together with the bundle around axis 1 that represents the central bundle line.

The pendulum with the two concentrated masses m_1 and m_2 can move in rotation around axis 2 and is supported by bearings. The pendulum is also connected to two cylindrical rubber materials that are protected by two tubes and fixed at the bottom of these ones. So, when the pendulum rotates around axis 2, there is a torsional deformation of the rubber material that causes an elastic restoring torque as well as dissipation of energy inside the rubber. It's the torsional damping effect of the TDD.

The whole device weights approximately 30 kg and has a gravity centre G_1 at about 35 cm below axis 1.

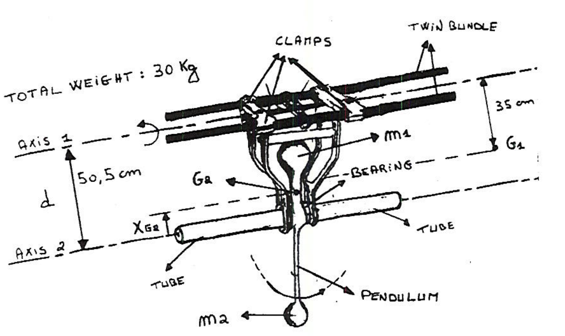

Fig. 1. Schematic representation of the TDD. Note that G_1 and G_2 are two different points.

So, for the whole line, the device acts also as a « pendulum » and thus increases the torsional stiffness of the latter. It's the detuning effect of the TDD.

B. Design

The design of the TDD has to be so that when the bundle begins to swing in torsion during a galloping event, the relative movement of the pendulum around axis 2 should be as big as possible to dissipate a maximum of energy in the rubber. The idea is thus to design the pendulum so that it possesses its own frequency in rotation around axis 2 as near as possible to the galloping frequency of the line where it must be installed. A schematic representation of the pendulum is given on Fig. 2.

δ = density
m_T = total mass
m_1 = 5.23kg
m_2 = 3.07kg
l_1 = 0.23m
l_2 = 0.5m
r_1 = 0.307m
r_2 = 0.565m
m_a = 0, 1, 2 or 2.5kg

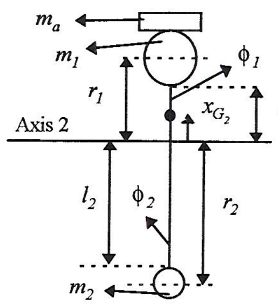

Fig. 2. Schematic representation of the pendulum

Its own frequency is given by

$$f_{pend} = \frac{1}{2\pi}\sqrt{\frac{K_{pend}}{I_{pend}}}$$
$$\text{with} \tag{3}$$
$$I_{pend} = m_1 r_1^2 + m_2 r_2^2 + \frac{\delta\phi_1 l_1^3}{3} + \frac{\delta\phi_2 l_2^3}{3}$$
$$K_{pend} = K_{int} - m_T g x_{G_2}$$

The meaning of the different symbols is obvious thanks to Fig. 2.

The galloping frequency of twin bundle conductors goes in general from 0.2 Hz to 0.6 Hz. The most difficult point in the design is to tune the pendulum on such low frequencies. Following formulas (3), we have to try to increase the inertia and to decrease the stiffness.

The big inertia explains why we have a big mass with a small arm above (because the place is limited) and a big arm with a small mass below (because the total mass of the system is limited). The stiffness term contains two parts : a part from the intrinsic stiffness of the rubber material and another one, function of the gravity centre position G_2 of the pendulum alone. In all cases we have to take care that the gravity centre of the pendulum remains above axis 2, otherwise, the stiffness would become too big. If now the gravity centre is too high, the pendulum will no longer swing around the vertical position that becomes unstable.

In order to range over the galloping frequencies, we can adjust the frequency of the pendulum with three degrees of freedom, namely :

1. Additional masses (m_a): the initial configuration of a pendulum is always with a gravity centre G_2 on axis 2. By adding some mass on the upper ball, it is possible to move the gravity centre G_2 above axis 2 with the effect of decreasing the stiffness.
2. Rubber diameter: this diameter can be decreased from 50mm to 40 mm leading also to a smaller stiffness.
3. Pendulum length: there are two lengths of pendulum that can be used to change the inertia.

IV. MANUFACTURING

The work to transfer the theoretical parameters of mass, stiffness and damping into a practical device was first imagined at the University of Liège, then manufactured by Dulmison and improved by Laborelec.

The expertise of a specialised chemist in rubber was used to determine the optimum elastomer for energy extraction properties. Initially the damping sleeve was made of an elastomer containing 50% natural rubber and 50% butyl rubber. This was subsequently changed to a 100% butyl compound, with improved damping properties.

In 1996, we are at the third generation of prototypes which now fulfils all requirements (electric, mechanic, easy to install, long life time, no interaction with aeolian vibration).

V. HIGH-VOLTAGE BEHAVIOUR

Laborelec (the Belgian Laboratory for Power Industry) conducted laboratory tests to assess the radio-electric disturbances and the noise level due to supplying high voltage to the TDD. Both dry and rainy conditions were investigated. The maximum voltage reached 242 kV, corresponding to 420 kV phase-to-phase on a transmission line. The measured disturbances on a bundle section equipped with one TDD are low in dry conditions and much higher in wet conditions, but do not exceed the usual allowed bad weather line emission threshold.

VI. DYNAMICAL BEHAVIOUR

A. Presentation

One prototype of the TDD was tested in Darmstadt in the laboratory of Professor Hagedorn. We chose a rubber diameter of 50mm. Others characteristics are summarised on Fig. 2. There are two main aims for testing such device in a laboratory: the first one is to assess the frequency range over which the damper works efficiently and the second one is to evaluate the damping capacity of the device.

A schematic view of the testing ground is given on Fig. 3.

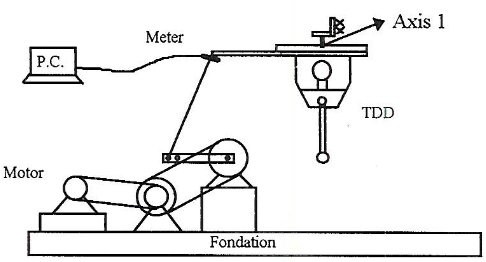

Fig. 3. Schematic view of the testing ground used in Darmstadt.

The whole system is excited in rotation around axis 1 that represents the central line of the bundle (on Fig. 3, axis 1 is pointing out of the paper). The excitation frequency varies between 0.2 Hz and 0.9 Hz with a step of 0.05 Hz. For each frequency, after the transient stage, we measure the applied force directly on a P.C. At the same time a video recording of the relative movement of the pendulum is made to evaluate the response in amplitude of the latter. For all the connections between the different mechanical elements we have used bearings to reduce the friction in the system to a minimum. The only important dissipation occurs in the rubber material of the TDD so that it becomes possible to estimate its damping characteristics. Four series of tests were realised with an additional mass respectively equal to 0, 1, 2 and 2.5 kg to change the position of the gravity centre G_2.

B. Results

The most relevant experimental results are summarised on Fig. 4. It is clear on this figure that the use of additional masses allows to reduce the resonance frequency from 0.65 Hz for $m_a = 0$ kg to 0.50 Hz for $m_a = 2$ kg. The width of each resonance peak is sufficient to say that each tested TDD can cover at least a frequency range of 0.2 Hz.

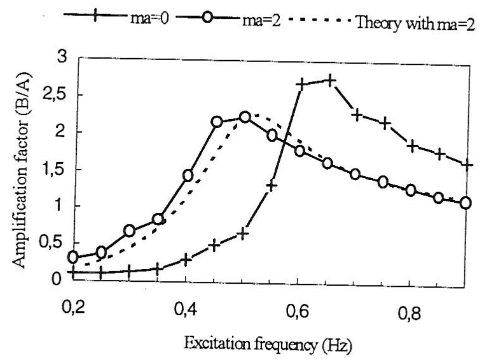

Fig. 4. Frequency response of the relative movement of the motion of the pendulum for two experimental cases (without additional mass, two kg more on the upper ball) and for the model fitting.

Two others factors have still to be tested in the future to reduce again the resonance frequency in order to cover the total range of galloping frequencies :

1. A rubber diameter smaller than 50 mm (decrease of the stiffness)
2. A longer pendulum (increase of the inertia)

The movement of the pendulum in the experimental conditions can also be simulated by the following equation if we assume small amplitudes of rotation:

$$I_{pend}\frac{d^2\theta_r}{dt^2} + 2\xi_{pend}w_{pend}I_{pend}\frac{d\theta_r}{dt} + \left(K_{int} - m_T g x_{G_2}\right)\theta_r = \left(I_{pend} - x_{G_2}m_T d\right)A\Omega^2 \sin(\Omega t) \qquad (4)$$

where θ_r is the relative rotation of the pendulum, w_{pend} is the own pulsation of the pendulum, A is the amplitude of the imposed oscillation around axis 1 and Ω is the pulsation of the imposed harmonic excitation. The damping is simply represented by a viscous term.

Let us define the response of the pendulum as:

$$\theta_r = B\sin(\Omega t - \phi) \qquad (5)$$

We are now able to calculate the amplitude of θ_r in function of the excitation frequency by replacing (5) in (4). After some developments, we have:

$$B = \frac{\left(I_{pend} - x_{G_2}m_T d\right)A\Omega^2}{K_{pend}\sqrt{\left(1 - \frac{\Omega^2}{w_{pend}^2}\right)^2 + \left(2\xi_{pend}\frac{\Omega}{w_{pend}}\right)^2}} \qquad (6)$$

and the maximum amplitude is reached for

$$\Omega_{max} = \frac{w_{pend}\sqrt{-1}}{\sqrt{2\xi_{pend}^2 - 1}} \qquad (7)$$

The model described by (5), (6) and (7) contains two unknown parameters, namely K_{int} and ξ_{pend}. We are now able to calibrate these two parameters with the experimental results. The calibration is made thanks to (6) and (7) so that the resonance frequency and the amplitude of the movement for the resonance frequency are respected for each series of test. This gives a system of eight equations for two unknown parameters so that we use a less square approximation. This calibration gives:

$$\begin{aligned} &K_{int} = 22.3\ Nm \\ &\xi_{pend} = 0.19 \text{ or } 19\% \text{ of the critical damping} \end{aligned} \qquad (8)$$

The dotted line plotted on Fig. 4 is the model fitting represented by (6) with the calibrated values calculated in (8) for the case $m_a = 2$kg.

VII. IMPACT ON THE WHOLE LINE

A. Detuning effect

The frequencies of the free oscillations obtained in (1) were derived for a horizontal twin bundled lines without TDD. If in the same case study we want to take into account the presence of N_{TDD} TDD distributed along the span, we have to correct the GJ, I and m terms in (1) by the following formulas [10]:

$$\begin{cases} GJ \rightarrow GJ + \left(\frac{L}{\pi}\right)^2 \frac{2}{L}\sum_{n=1}^{N_{TDD}} m_{TDD} g l_{TDD} \sin^2\left(\frac{z_{TDD,n}\pi}{L}\right) \\ I \rightarrow I + \frac{2}{L}\sum_{n=1}^{N_{TDD}} m_{TDD} l_{TDD}^2 \sin^2\left(\frac{z_{TDD,n}\pi}{L}\right) \\ m \rightarrow m + \frac{2}{L}\sum_{n=1}^{N_{TDD}} m_{TDD} \sin^2\left(\frac{z_{TDD,n}\pi}{L}\right) \end{cases} \qquad (9)$$

where m_{TDD} is the total mass of the TDD (30 kg), l_{TDD} is the distance between the central bundle line and G_1 (0.35 m),

$z_{TDD,n}$ is the inspan location of the n[th] TDD and g is the gravity constant.

The detuning effect of the TDD is obvious on Fig. 5.

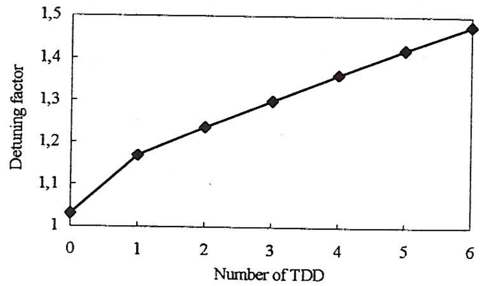

Fig. 5. Detuning factor ($\frac{f_\theta}{f_v}$) in function of the number of TDD if they are regularly spaced along the span.

B. Damping effect

The percentage of critical damping in torsion is generally assumed to be around 4% for a horizontal twin bundled lines without TDD.

With the same hypothesis as the ones given in section 2 for a horizontal twin bundled line, and thanks to an energy balance method, we are able to calculate the increase of the percentage of critical damping in torsion introduced by N_{TDD} distributed TDD along the span :

$$\xi_\theta \to 0.04 + \frac{2\xi_{pend}}{L}\,\frac{w_{pend}}{w_\theta}\,\frac{I_{pend}}{I}\,H^2(w_{gal})\left[\sum_{n=1}^{N_{TDD}} \sin^2\left(\frac{z_{TDD,n}\pi}{L}\right)\right] \tag{10}$$

where $H(w_{gal})$ is the amplification function evaluated in laboratory (B/A). This is a function of the excitation pulsation w_{gal} which is for a bundled line, the galloping pulsation.

On Fig. 6, we can see that with 3 TDD (ma=2kg), the torsional damping is nearly doubled (>0.07) around 0.5 Hz.

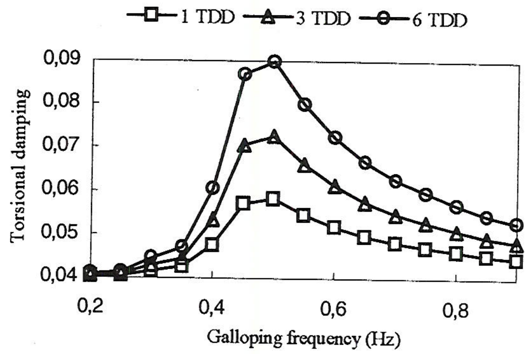

Fig. 6. Percentage of critical damping in torsion in function of the galloping frequency for 1, 3 or 6 TDD (ma=2kg) uniformly distributed along the span.

VIII. FIELD RESULTS

These observations occurred on a Belgian 380 kV transmission line, between two dead-end towers. The phases are made of twin bundled AMS 620 mm^2 conductors. The same span includes an experimental phase, identical, parallel and next to the operational ones, but equipped with two TDD prototypes. These were tuned to the most probable galloping frequencies. Load sensors are used for tension recording at the phases attachments to the first tower. One case of large amplitude tension variations was detected on 17/11/1992 by the recording station.

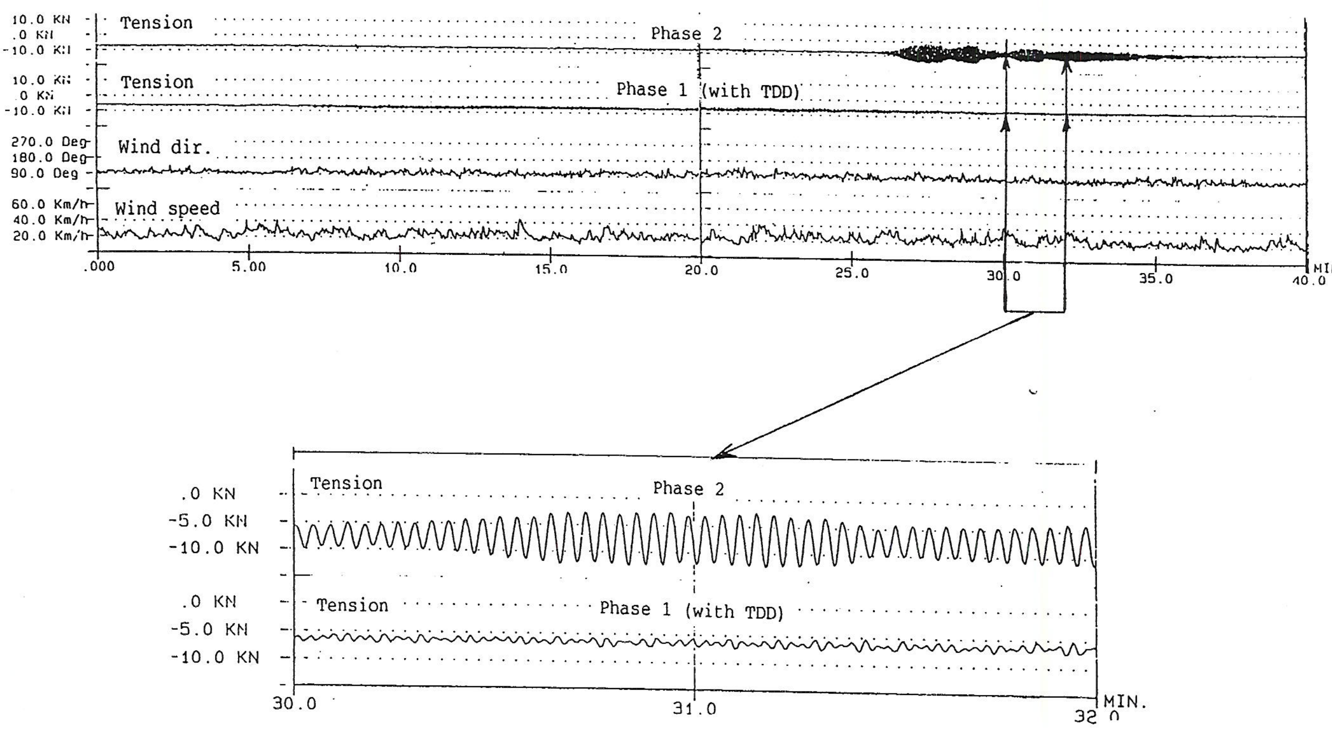

Fig. 7. Time diagram of the galloping record in Villeroux: phase 1 is equipped with two TDD

The results lead to the conclusion that the amplitudes of the tension variations measured on the TDD protected phase are considerably lower (7 times) than those of the non-protected ones, due to the efficiency of the TDD's.

The time diagram on Fig. 7 clearly shows how the TDD really acts: from minute 5 to 25, the phase without TDD does not move while a regular but very limited vibration is observed on the phases with TDD. This is the consequence of the TDD resonance: this TDD movement absorbs most of the inputted energy and prevents the wind energy transfer to vertical motion. On the contrary, from minute 26 to 35, a large amplitude vibration develops on the non-protected phases, while the light movement of the protected phase progressively disappears.

IX. CONCLUSIONS

A damping system for galloping of bundled lines has to combine a **detuning effect** with a **high torsional damping** to ʼoid the instability.

The anti-galloping device presented in this paper, called TDD for Torsional Damper and Detuner satisfies these two criteria as well as all the necessary requirements of overhead transmission lines like low radio-electric disturbances and noise level, long life time, no interaction with aeolian vibration and facility for installation.

Moreover the transposition of the laboratory results to the whole line confirms the detuning effect as well as the high torsional damping introduced by the TDD.

Finally the design of the TDD allows, thanks to the use of different additional masses, different rubber diameter and different pendulum length, to cover the total range of galloping frequencies.

X. ACKNOWLEDGEMENTS

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XII. BIOGRAPHIES

Renaud Keutgen was born in Liège, Belgium, on October 18, 1971. He received his degree in Mechanical Engineering from the University of Liège in 1994. He is currently a Ph.D. student at the same University. In 1996, he spent two months in the department of Professor Hagedorn in Darmstadt, Germany.

Jean-Louis Lilien was born in Liège, Belgium, on May 24, 1953. He received his degree in Electrical and Mechanical Engineering from the University of Liège in 1976. He received his Ph.D. from the same University in 1984. He is presently a professor at the same University, Dept. of Transmission and Distribution of Electrical Energy. His main activity is based on the effects of short-circuit mechanical effects and overhead lines vibrations (galloping). He is the chairman of the CIGRE task force on the effects of short-circuit in substation (belonging to WG 23-11). He is also a expert of the CIGRE task force on galloping (belonging to WG 22-11). He has published over 60 technical papers and participated to many symposia and international conferences. He received the international prize « George Montefiore » in 1986.