Acta Mechanica Slovaca 13 (3): 38-47, 2009
DOI: 10.2478/v10147-010-0045- x

Optimal Tuning Method of Ships System by Means of Pneumatic Tuner of Torsional Oscillations

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ABSTRACT

Drives of mechanical systems consist of driving machines and gearing mechanisms connected together by means of shafts and couplings. With regard to the dynamics it is possible to say that every driving mechanism is a system, which is able to oscillate. Especially piston machines are able to create dangerous and harmful torsional oscillations, vibrations, as well as noisiness.

Important task of designer is reduction of torsional oscillations in mechanical system. Presently this problem is solved predominately with a suitable adjustment of dynamic properties of flexible shaft couplings with regard to the dynamic properties of given system. It means that every torsionally oscillating mechanical system needs to be tuned suitably. The main purpose of this paper is presentation of controlling possibilities of dangerous torsional oscillations of mechanical systems by means of a new method, i.e. its optimal tuning by means of pneumatic coupling with self-regulation, developed by us.

KEYWORDS

dangerous torsional oscillations, tuning of mechanical system, behaviour of optimal tuning of the system, pneumatic coupling with self-regulation - pneumatic tuner.

INTRODUCTION

Pumps, fans and especially piston combustion engines, as well as compressors, are exciters or actuators of torsional oscillations in mechanical systems. Intensive torsional oscillations are the main cause of excessive dynamic load in these systems. As a result of this fact, there are rising damages or failures of individual parts in the system, for example damages of bearings, shafts, transmissions and flexible shaft couplings or even destructions of piston machinery.

The successful control of torsional oscillations can be achieved only after a detailed analysis of the system from the torsional dynamic point of view. The results of such analysis indicate that stated goal is possible to be achieved only when the values of some elements from the system will be adjusted suitably to the dynamic of the system. That means, that any ”torsionally oscillating mechanical system” (TOMS) has to be tuned. Practical experiences shown that the most suitable medium for tuning of any TOMS is the properly modified flexible shaft coupling.

In our department, besides other activities, we are researching and developing flexible pneumatic shaft coupling, particularly the differential pneumatic flexible shaft coupling with the self-regulation, which is working in the given mechanical system as a ”pneumatic tuner of torsional oscillations” (PToTO). We are focusing our attention, as of now only in theoretical level, to the application possibilities of given coupling in the TOMS.

The aim of this paper is to inform the technical community about possibilities of control of dangerous oscillations of mechanical systems by means of a new method, i.e., using application of pneumatic coupling with the self-regulation, developed by us and also to present the calculation of equations for optimal tuning of the system.

THE CHARACTERISTIC OF TORSIONAL OSCILLATION OF MECHANICAL SYSTEMS AND POSSIBILITIES OF ITS CONTROL

The theory of oscillations defines that the n-mass mechanical system has $(n - l)$ natural frequencies of torsional oscillations, as well as the same number of corresponding forms of oscillations. To each of these oscillation forms corresponds a certain range of critical speeds, which are the result of resonance of harmonic parts of actuating (load torque - LT) of piston machinery with the natural frequency corresponding to the form of oscillations of the system. In the case of failure-free operation of the piston device during the lowest forms of oscillations (one-nodal, two-nodal), the most dangerous are the revolutions actuated by the main harmonic part of actuating of piston device, which we call the main critical revolutions. The desired position is in the case of performance of machinery in above-resonance level, the main critical revolutions were positioned under the non-load revolutions, or at least, in 40 % -distance from the minimal revolutions of running regime of the system. Then, in the range of running regime of the system, the remaining are so-called secondary critical revolutions, in which the level of torsional actuating is small. The level of the torsional actuating of secondary critical revolutions will increase, if occurs a damage of the piston machinery. As a consequence of unequal actuating of individual pistons of piston machinery in range of working regime of the system, occurs a very intensive resonance arising from lower harmonic parts of actuating [1], [18], [19]. Dangerous torsional oscillation can be controlled by menas of detailed analysis of the system from the torsional dynamic point of view.

This problem is relevant for every manufacturer of flexible shaft couplings. Worldwide well-known manufacturers are trying to solve this problem by utilizing of the highly flexible shaft couplings, that is, the couplings with the high torsional rigidity [1], [18], [17], [20], [21].

Our department is also involved into solving of this problem. We are trying to control the dangerous torsional oscillations by application of the pneumatic flexible shaft couplings, in particular by utilization of the differential pneumatic flexible couplings as the PToTO, developed by us [6], [7]

By means of utilization of the pneumatic couplings we are able to control the dangerous torsional oscillations and in this way to ensure tuning of the TOMS during the running regime of the system [13].

The basic task of tuning of the TOMS is a suitable adjustment of dynamic properties of pneumatic coupling within running regime of the TOMS [9], [10], [13].

Furthermore we are focusing our attention to the possibilities dangerous torsional oscillations control in a real ships driving system during the current operation, by means of application of the pneumatic differential coupling in function of the PToTO.

BRIEF CHARACTERISTICS OF DIFFERENTIAL PNEUMATIC COUPLINGS AND CONTINUOUS TUNING OF MECHANICAL SYSTEM

Pneumatic differential couplings, according to the Fig. 1 and Fig. 2 have to fulfil next important requirements:

a) compensation of axial, radial and angular misalignments, caused due to manufacturing inaccuracies,

b) to keep stable dynamic mechanical properties and constant flexible transfer of loading torque during the whole technical life of the mechanical system,

c) ability to tune torsionallly oscillating mechanical systems continuously.

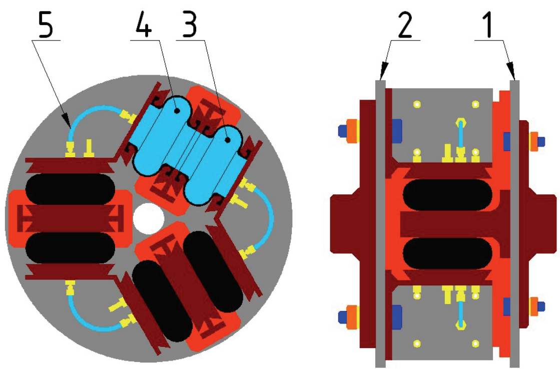

Fig. 1: *Scheme of the pneumatic flexible differential shaft coupling.*

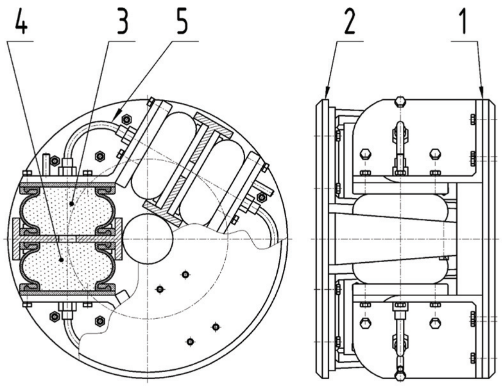

Fig. 2: *Real assembly of the pneumatic flexible differential shaft coupling.*

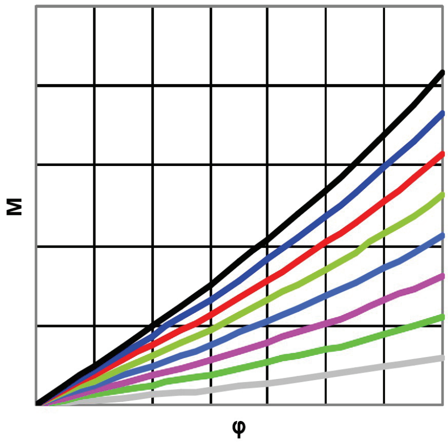

Fig. 3: *Behaviours of static characteristics of pneumatic differential coupling for pressures of gaseous medium* $p = 100 \div 800$ *kPa.*

Pneumatic coupling, see Fig. 1 and Fig. 2, consists of the driving part (1) and the driven part (2), between them is a compression space filled with a gaseous medium. The compression space consists of three circumferentially dislocated, differential components, connected each other with the pipes (5). Every differential element has the compressed (3) and the pulled (4) pneumatic-flexible element.

By means of change of the gaseous medium pressure (p) in the compression space of pneumatic coupling, it can be changed characteristics of the coupling (Fig. 3), as well as there can be changed characteristic mechanical properties of the coupling (torsional rigidity and dumping coefficient). In this way there is changed, i.e. tuned, dynamic torsional rigidity (Fig. 4), which is the most important factor of natural frequency in the given system $\Omega = \sqrt{k/I_{red}}$. According to this fact it is evident the basic principle of continuous tuning of the TOMS by means of pneumatic couplings. The natural angular frequency of system (Ω) is modified with regard to the actuating angular frequency (ω), in order to avoid the resonance state ($\Omega = \omega$) or state very closed to the resonance phenomenon.

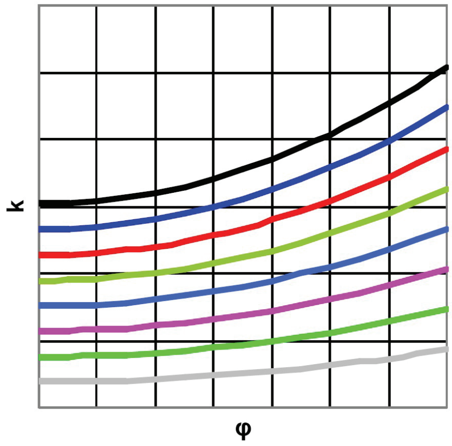

Fig. 4: *Behaviours of static characteristics of pneumatic differential coupling for pressures of gaseous medium* $p = 100 \div 800$ *kPa.*

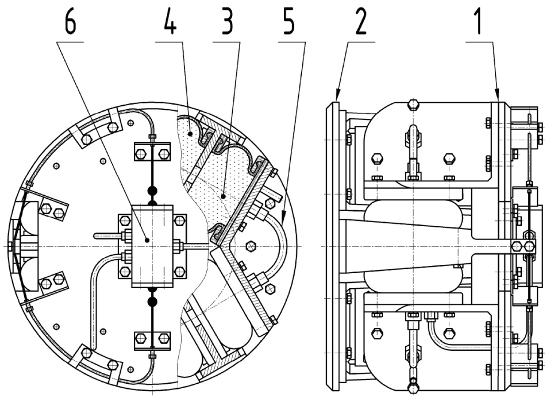

Fig. 5: *Pneumatic tuner of torsional oscillations.*

Application of continuous tuning, according to the invention [5], requires also new application of another kind of coupling. It is coupling, i.e. the

PToTO, which is able to change its basic characteristics, like torsional rigidity and damping coefficient. This requirement is fulfilled in the case of pneumatic differential coupling, which is developed newly by us and which is able to operate in the function of torsional oscillations tuner (Fig. 5).

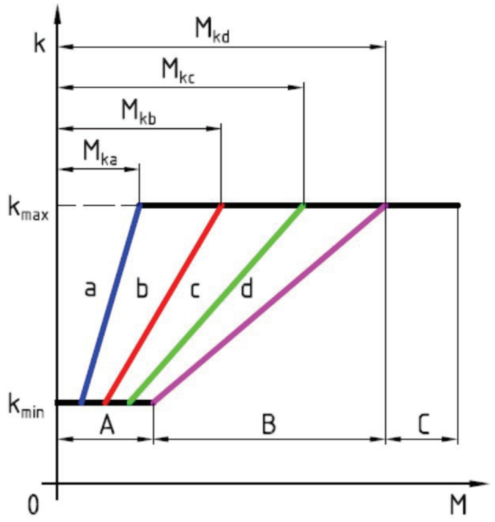

Fig. 6: *Behaviour of torsional rigidity k of pneumatic tuner of torsional oscillations in dependence on loading torque M.*

Pneumatic tuner of torsional oscillations (Fig. 5), protected by two patents for an invention [10], [12], is similar to the pneumatic differential coupling. The main difference consists in regulator (6), which enables to keep constant angle of twist in the coupling. The basic principle of the PToTO is the self-regulation ability of the angular twisting, caused due to actual change of loading torque, into the given constant angular value φ_k. This self-regulation of gaseous medium pressure in the compression space in the tuner has an immediate influence onto the characteristic of pneumatic tuner (Fig. 3) and, of course, onto the torsional rigidity k (Fig. 6). In this way it can be changed the natural frequency of the system.

There are presented behaviours of torsional rigidity of the PToTO on the Fig. 6 in dependence on the loading torque. To each of the calculated constant angles of twist φ_{k1}, φ_{k2}, φ_{k3} and φ_{k4} belongs one behaviour of torsional rigidity a, b, c, d.

The above-mentioned behaviours are limited with minimum and maximum values of torsional rigidity k_{min} and k_{max} according to the pressures of gaseous medium from interval p_{min} and p_{max} in compression space of the PToTO. There are also presented behaviours illustrated by a fractional line, which consists of three areas: pre-regulation - A, regulation - B and over-regulation - C area. From this illustration it is evident that change of φ_{ki} influences interval of pre-regulation and regulation area, but it influences mainly the value of pneumatic tuner torsional rigidity during operational regime of the system. There are also influenced values of loading torque M_1, M_2, M_3 and M_4 with regard to the maximum value k_{max} of torsional rigidity. The pneumatic tuner with an increasing value of constant angle of twist, during a certain loading torque, has a declining torsional rigidity. In the case of the PToTO with the maximum angle of twist value φ_{kmax}, from the relatively hard pneumatic coupling (behaviour a) becomes a high flexible pneumatic coupling (behaviour d), which is able to operate with considerably higher value of loading torque M_d at maximum value of dynamic torsional rigidity.

CHARACTERISTICS OF TORSIONALLY OSCILLATING MECHANICAL SYSTEM

The described torsionally oscillating mechanical system (Fig. 7) consists of the driving part (1), driven part (2), PToTO (4), gearbox (3), flexible coupling (5) and shaft (6). The driving part is the six-cylinder in-line Diesel engine - type $6-27,5A2LS$, with power output $P = 515$ kW and speed range $n = 200 \div 600$ min^{-1}. The driven part is a three-vane propeller with diameter 1700 mm. The transmission of torque between the driving and driven parts consists of the PToTO - type $3-1/210-D/A$, reverse gearbox - type $VSR10B$ (3) with gear ratio $i = 1,766$, flexible coupling with rubber band - type $VS600$ and shafts.

This driving unit is installed in a real ships system, namely in a river-push boat, Fig. 8.

LOADING ANALYSIS OF PNEUMATIC TUNER OF TORSIONAL OSCILLATIONS

Analysis of loading of the PToTO during a steady state operation of mechanical system will be per-

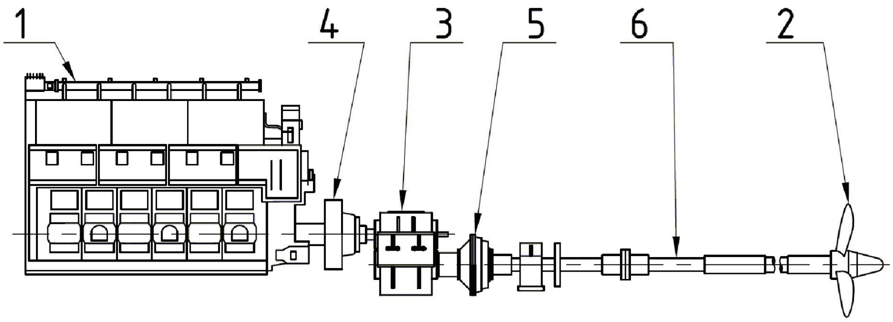

Fig. 7: *The real torsionally oscillating mechanical system.*

Fig. 8: *River-push boat.*

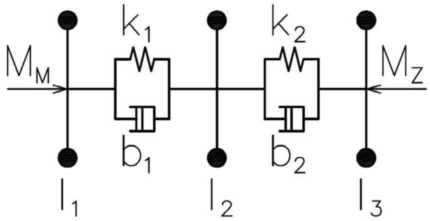

Fig. 9: *Schematic model of given torsionally oscillating mechanical system.*

formed by means of a schematic model of torsionally oscillating mechanical system (Fig. 9).

During the calculation process of mechanical system loading during the steady state operation in the range of its working regime, it can be supposed that this mechanical system is rotating with angular speed ω, which is changing in the framework of working regime. The mass (1) with moment of inertia I_1 is loaded by loading torque in the form

$$M_z + \sum M_j \sin(j\omega t + \gamma_j).$$

From this relation it is evident that the PToTO, as well as the whole torsionally oscillating mechanical system, is loaded by loading torque with fan characteristics, i.e. $M_z = 0,043n^2$ and there is also added a harmonic component M_j of oscillation. According to this, the PToTO has to transmit also additional component of dynamic moment M_d. So, in our case, the pneumatic tuner is loaded with loading torque M_K and maximum angle of twist φ_K:

$$M_K = M_Z + M_d, \tag{1}$$

$$\varphi_K = \varphi_Z + \varphi_d. \tag{2}$$

The additional dynamic moment M_d and the dynamic component of maximum angle of twist φ_d can be calculated from equations of motion (3).

Equations of motion for the three-mass mechanical system:

$$\begin{aligned} &I_1\ddot{\varphi}_1 + b_1(\dot{\varphi}_1 - \dot{\varphi}_2) + k_1(\varphi_1 - \varphi_2) = M_M(t) \\ &I_2\ddot{\varphi}_2 - b_1(\dot{\varphi}_1 - \dot{\varphi}_2) - k_1(\varphi_1 - \varphi_2) \\ &\qquad\qquad + k_{12}(\varphi_2 - \varphi_3) = 0 \\ &I_3\ddot{\varphi}_3 - b_2(\dot{\varphi}_2 - \dot{\varphi}_3) - k_2(\varphi_2 - \varphi_3) = -M_Z(t), \end{aligned} \tag{3}$$

whereas the loading torque of engine is described in a complex form (4):

$$M_M = M_Z + M_d = M_z + \sum_{j=1}^{n} M_j e^{i(\omega j t - \Delta\varphi_{Mj})}, \tag{4}$$

where

M_Z – is loading torque, this moment is stationary in steady state of the system,

M_d – is component of additional engine's dynamic moment,

M_j – is amplitude of j-harmonic component of additional engine's dynamic moment,

$\Delta\varphi_{Mj}$ – is phase of j-harmonic component of additional engine's dynamic moment.

Natural speed frequencies of this system are:

$$N_{1,2} = \frac{\Omega_{1,2}60}{2\pi}, \tag{5}$$

where:
$\Omega_{1,2}$ – are natural angular frequencies of the system, according to the relation (6).

$$\Omega_{1,2} = \sqrt{\frac{q_2 \mp \sqrt{q_2^2 - 4q_1q_3}}{2q_1}}, \tag{6}$$

where
coefficients q_1, q_2 a q_3 are calculated from the next equations:

$$q_1 = \frac{I_1I_2I_3}{k_1k_2}, \tag{7}$$

$$q_2 = \frac{(I_2+I_3)I_1}{k_1} + \frac{(I_1+I_2)I_3}{k_2}, \tag{8}$$

$$q_3 = I_1 + I_2 + I_3. \tag{9}$$

For calculation of enforced torsional oscillation it is meaningful to apply a suitable substitution for torsion of *coupling 1*: $\varphi_{S1} = \varphi_1 - \varphi_2$ and for torsion of *coupling 2*: $\varphi_{S2} = \varphi_2 - \varphi_3$, as well as for stationary moment M_Z and, in this way, to modify system of equations of motion (3) into:

$$\begin{aligned} &\ddot{\varphi_{S1}} + b_1\left(\frac{I_1+I_2}{I_1I_2}\right)\dot{\varphi_{S1}} + k_1\left(\frac{I_1+I_2}{I_1I_2}\right)\varphi_{S1} \\ &\quad - \frac{b_2}{I_2}\dot{\varphi_{S2}} - \frac{k_2}{I_2}\varphi_{S2} = \frac{M_d(t)}{I_1}, \\ &\ddot{\varphi_{S2}} + b_2\left(\frac{I_2+I_3}{I_2I_3}\right)\dot{\varphi_{S2}} + k_2\left(\frac{I_2+I_3}{I_2I_3}\right)\varphi_{S2} \\ &\quad - \frac{b_1}{I_2}\dot{\varphi_{S1}} - \frac{k_1}{I_2}\varphi_{S1} = 0. \end{aligned} \tag{10}$$

Behaviour of harmonic components of angle of twist, as well as dynamic moment of engine in a complex form is:

$$\varphi_d = \sum_{j=1}^{n} \varphi_{Skj} e^{i(\omega j t - \Delta\varphi_{Skj})}, \tag{11}$$

$$M_d = \sum_{j=1}^{n} M_j e^{i(\omega j t - \Delta\varphi_{SMj})}, \tag{12}$$

where

φ_{Skj} – is amplitude of dynamic angle of twist of k-coupling for j-harmonic component,

$\Delta\varphi_{Skj}$ – is phase of torsion of k-coupling for j-harmonic component of dynamic engine's moment.

Next there is created a system of two equations for both harmonic components and they can be solved using Cramer's rule.

For amplitudes of individual harmonic components of torques and angles of twist there are given equations resulting from the presented solution:

$$M_{Skj} = \sqrt{k_k^2 + b_k^2 \omega^2 j^2} \varphi_{Skj}, \tag{13}$$

where

M_{Skj} – is amplitude of dynamic moment of k-coupling for j-harmonic component.

$$\varphi_{S1j} = \frac{M_j}{I_1(D_R^2 + D_I^2)} \sqrt{K^2 + L^2}, \tag{14}$$

$$\varphi_{S2j} = \frac{M_j}{I_1(D_R^2 + D_I^2)} \sqrt{M^2 + N^2}. \tag{15}$$

The required coefficients can be obtained from given relations:

$$\begin{aligned} K &= G \cdot D_R + H \cdot D_I, \\ L &= -G \cdot D_I + H \cdot D_R, \\ M &= E \cdot D_R + F \cdot D_I, \\ N &= -E \cdot D_I + F \cdot D_R; \end{aligned} \tag{16}$$

$$\begin{aligned} D_R &= (A \cdot G - B \cdot H - C \cdot E + D \cdot F), \\ D_I &= (A \cdot H - B \cdot G - C \cdot F + D \cdot E); \end{aligned} \tag{17}$$

$$\begin{aligned} A &= k_1 \frac{I_1 + I_2}{I_1 I_2} - \omega^2 j^2, \\ B &= b_1 \frac{I_1 + I_2}{I_1 I_2} \omega j, \\ C &= -\frac{k_2}{I_2}, \\ D &= -\frac{b_{2j}}{I_2} \omega j, \\ E &= -\frac{k_1}{I_2}, \\ F &= -\frac{b_1}{I_2} \omega j, \\ G &= k_2 \frac{I_2 + I_3}{I_2 I_3} - \omega^2 j^2, \\ B &= b_2 \frac{I_2 + I_3}{I_2 I_3} \omega j. \end{aligned} \tag{18}$$

THEORETICAL RESULTS OF OPTIMAL TUNING OF SHIPS SYSTEM

According to theoretical results concerning torsional oscillation, we can evaluate the tuning process of the ships system during operation in a steady state with regard to application of the pneumatic tuner of torsional oscillations.

Torsional oscillation of given system will be presented by:

- behaviours of amplitudes of dynamic moment caused by torsional oscillation of flexible couplings depending on speed,
- behaviours of dynamic component of angle of twist amplitudes depending on speed.

Influence of the PToTO on the torsional oscillation of ships system was verified on condition of equal actuating of individual pistons in engine.

There is presented on the Campbell's diagram (Fig. 10) a tuning process of the ships system, which is realized by means of an application of the PToTO in the steady state. On this figure is illustrated a group of eight behaviours of the first natural speed frequencies, with designation a_1, b_1, c_1, d_1, e_1, f_1, g_1, h_1 and here is also a group of eight behaviours of the second natural speed frequencies, designated a_2, b_2, c_2, d_2, e_2, f_2, g_2, h_2, which are corresponding to the constant angles of twist in the PToTO in the range $\varphi_K = 1°; 2°; 3°; 4°; 5°; 6°; 7°$ and 8°.

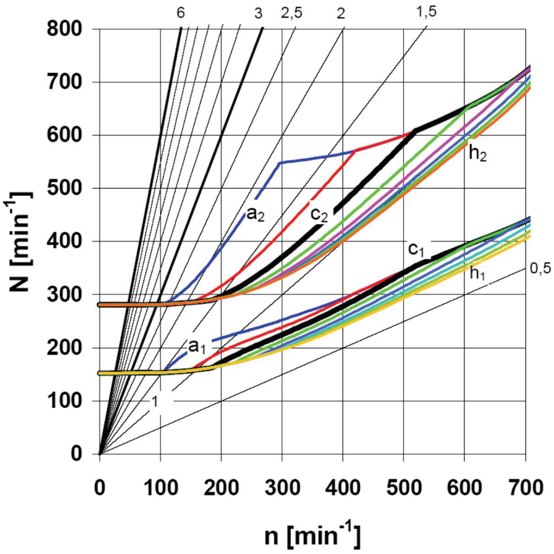

Fig. 10: *Campbell's diagram of given ships system.*

In accordance with the Campbell's diagram it can be declared that the ships system is tuned optimally in the range of operational speed ($n = 200 \div$

700 min^{-1}) with the constant angle of twist value of the PToTO $\varphi_K = 3°$. This fact is presented by the behaviour c_1 for the first speed frequency and by the behaviour c_2 for the second speed frequency. The behaviour c_1 passes between the half ($j = 0,5$) and first ($j = 1$) harmonic component in interval of the operational speed, whilst the behaviour c_2 passes between the first ($j = 1$) and first and half ($j = 1,5$) harmonic com-ponent of loading torque. It means that in the range of operational speed there is no cross-point among the given behaviours (c_1 and c_2) and the lower harmonic components, i.e. there is no danger of critical speed occurrence. Based on the above-mentioned facts it is possible to say that there is no risk of any dangerous torsional oscillation in the range of the operational speed. This fact is supported by the figures Fig. 9 and Fig. 10. They illustrate tuning of the ships system using the PToTO with the constant angles of twist $\varphi_K = 3°$. From these figures it can be seen that for the speed values from the interval $n = 0 \div 700$ min^{-1} it is increased amplitude of the dynamic moment M_d. The given figures describe also torsional oscillation of the system by means of behaviours of the dynamic moment M_d amplitudes and dynamic angle of twist φ_d amplitudes near the speed values $60 \div 80$ min^{-1}, what is a resonance area with the main harmonic component ($j = 3$) from the first and the second natural speed frequencies. It corresponds, in the concrete, to the next loading state of the PToTO: $M_d = 3071,62$ Nm and $\varphi_d = 9,26°$, whilst the loading of flexible coupling $VS600$ is: $M_d = 2889,32$ Nm and $\varphi_d = 3,03°$. If the operational speed is more far away from the resonance area, the loading of individual flexible couplings descends, what is favourable situation for the whole ships system. Loading of the individual flexible couplings in the range of operational speed $n = 200 \div 700$ min^{-1} is as follows:

- in the PToTK: $M_d = 211,36 \div 131,54$ Nm, $\varphi_d = 0,54 \div 0,38°$,
- in the flexible coupling $VS600$: $M_d = 49,57 \div 26,11$ Nm, $\varphi_d = 0,0550,0058°$.

According to the all above-mentioned results it is possible do declare that by means of application of the PToTK with constant value of angle of twist $\varphi_K = 3°$ we are able to achieve a smooth operation of the ships system with regard to its torsional oscillation.

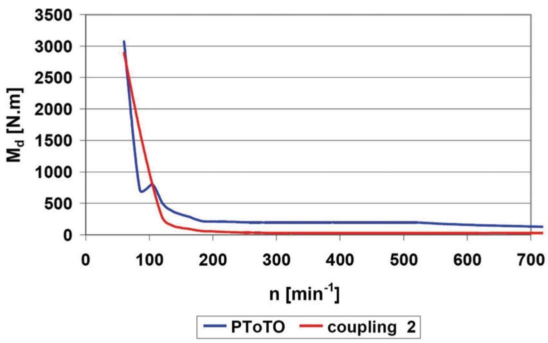

Fig. 11: *Behaviours of amplitudes of dynamic moment M_d depending on the speed n for the given ships system.*

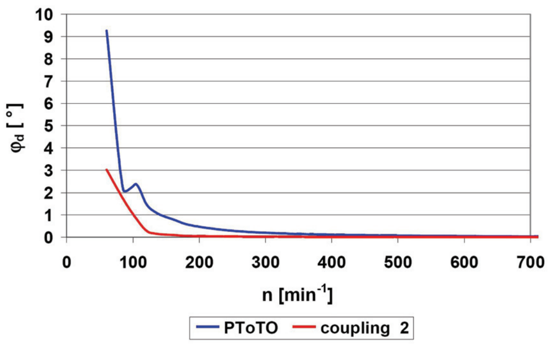

Fig. 12: *Behaviours of amplitudes of angle of twist dynamic component φ_d in coupling depending on the speed n for the given ships system.*

CONCLUSION

Results from this experimental work confirmed also an important fact that the dangerous torsional oscillations in mechanical system can be reduced into an acceptable level by means of a suitable modification of dynamic properties of flexible couplings in this system. Torsionally oscillating mechanical systems have to be tuned in advance. For proper tuning of torsionally oscillating mechanical systems it is necessary to perform detailed dynamic calculation with regard to the torsional oscillation. Taking into consideration the above-mentioned facts, our suggestion is: to apply pneumatic flexible shaft couplings developed by us for tuning of the ships system in order to reduce dangerous torsional oscillations. These flexible shaft couplings are in fact pneumatic tuners of torsional oscillations. They have not only one, but the whole range of characteristics and characteristic features in the framework of gaseous medium pressure in the compression area with regard to the chosen constant value of angle of twist.

During application of pneumatic tuner of torsional oscillations with the constant angle of twist $\varphi_K = 3°$ was not occurred a resonance phenomenon in the whole range of operational regime because the pneumatic tuner fulfils also a function of high-flexible shaft coupling after the change of constant angle of twist.

It is possible to say, finally, that this tuning method during current operation (continuous tuning of systems) can be applied in every situation where should be eliminated dangerous torsional oscillation of mechanical systems. The results presented in this paper confirm our presumption that these new tuning methods are able to improve technical standard and operational reliability in all torsionally oscillating mechanical systems.

ACKNOWLEDGMENT

This paper was written in the framework of Grant Project VEGA: "1/0304/09 – Governing of dangerous vibrations in drives of mechanical systems".

REFERENCES

[1] Böhmer J., Einsatz elastischer Vulkan-Kupplungen mit linearer und progressiver Drehfedercharakteristik, MTZ Motortechnische Zeischritt, 44 Jahrgang, No. 5, 1983.

[2] Grega R., Prezentácia výsledkov dynamickej torznej tuhosti pneumatickej pružnej spojky s autoreguláciou na základe experimentálnych meraní, "Optimalizcia mechanickch sstava zariaden s aplikciou prvkov ast strojov", Acta Mechanica Slovaca, vol. 6, no. 2, 2002, p. 29–34.

[3] Grega R., Optimalizačný nomogram minimalizácia rozmerov pneumatickej pružnej spojky, Acta Mechanica Slovaca, vol. 10, no. 4-B, 2006, p. 35–38.

[4] Grega R., Výsledky experimentálneho overovania statickej optimalizácie v torzne kmitajúcej mechanickej sústave, Acta Mechanica Slovaca, vol. 11, no. 4-A, 2007, p. 21–26.

[5] Homišin J., Mechanická sústava vhodná pre realizáciu plynulého ladenia, Patent no. 276926/92.

[6] Homišin J., Pneumatická pružná hriadeľová spojka so schopnosťou autoregulćie (Pneumatic flexible shaft coupling with ability of self-regulation), Patent no. 278025/95.

[7] Homišin J., Pneumatická spojka s prídavným regulátorom konštantného uhla skrútenia (Pneumatic coupling with additional regulator of constant twist angle), Patent no. 278272/96.

[8] Homišin J., Methods of tuning torsionally oscillating mechanikal systems using pneumatic tuners of torsional oscillations, Transactions of the Technical university of Košice, vol. 3, no. 4, 1993, p. 415–419.

[9] Homišin J., Aplikácia diferenčných pneumatických spojok s autoreguláciou a bez autoregulácie v torzne kmitajúcich mechanických sústavách, Strojnícky časopis, vol. 48, no. 2, 1997, p. 116–125.

[10] Homišin J., Aplikácia diferenčnej pneumatickej spojky s prídavnm regulačným obvodom, Strojnícky časopis, vol. 49, no. 2, 1998, p. 106–111.

[11] Homišin J., Newly-developed pneumatic clutches functioning as the tuner of torsionally oscillating mechanical systems, Part 1. Tuning-up of modelledtorsionally oscollating mechanical systems via differential pneumatic flexible clutch, Transactions of the Technical university of Košice, vol. 2, 1998, p. 47–52; Part 2. Tune-up of modelled torsionally oscollating mechanical systems via differential pneumatic flexible clutch eith the autoregulation, Transactions of the Technical university of Košice, vol. 2, 1998, p. 53–57.

[12] Homišin J., Jurčo M., Aplication of differential pneumatic couplings voith and without autoregulation in torsionally oscillating mechanical systems, The shock and vibration digest, vol. 29, no. 3, 1997, 44 p.

[13] Homišin J., Jurčo M., Application of differential pneumatic clutch with an additional regulating system, The shock and vibration digest, Virginia Tech., vol. 30, no. 6, 1998, 490 p.

[14] Kaššay P., Algoritmus extremálnej regulácie s redukciou kroku a jeho overenie v nasimulovanej torzne kmitajúcej mechanickej sústave, Acta Mechanica Slovaca, vol. 10, no. 4-B, 2006, p. 49–54.

[15] Kaššay P., Stanovenie štartovacieho bodu pre extremálnu reguláciu torzne kmitajúcich mechanických sústav, Rozprawy naukowe, ATH, Bielsko-Biaa, PL, no. 25, 2008, p. 79–83.

[16] Kaššay P., Experimentálne overenie možnosti realizácie extremálnej regulácie v torzne kmitajúcich mechanických sstavách, Acta Mechanica Slovaca, vol. 12, no. 3-C, 2008, p. 53–58.

[17] Lunke M., Beefting G.B., Einsatz Hochelastisher Kupplungen in energiesparenden Schiffsantriebsanlangen, Schiff und Hafel, vol. 4, no. 35, 1983.

[18] Zoul V., Některá hlediska vývoje pružných spojek pro soustrojí s naftovými motory, Strojírenství, vol. 32, no. 6–7, 1982.

[19] Zoul V., Vliv nevyrovnaného buzení jednoltivých válců naftového motoru na torzní kmitání soustrojí, Strojírenství, vol. 33, no. 6–7, 1983.

[20] Zoul V., Vysokopružné spojky RATO, Informační spravodaj, vol. 20, ČKD Praha, 1988.

[21] Zoul V., Použití pružných hřídelových spojek s nízkou torzní tuhostí k snížení dynamického torzního namáhaní, Informační spravodaj, vol. 24–25, ČKD Praha, 1989.